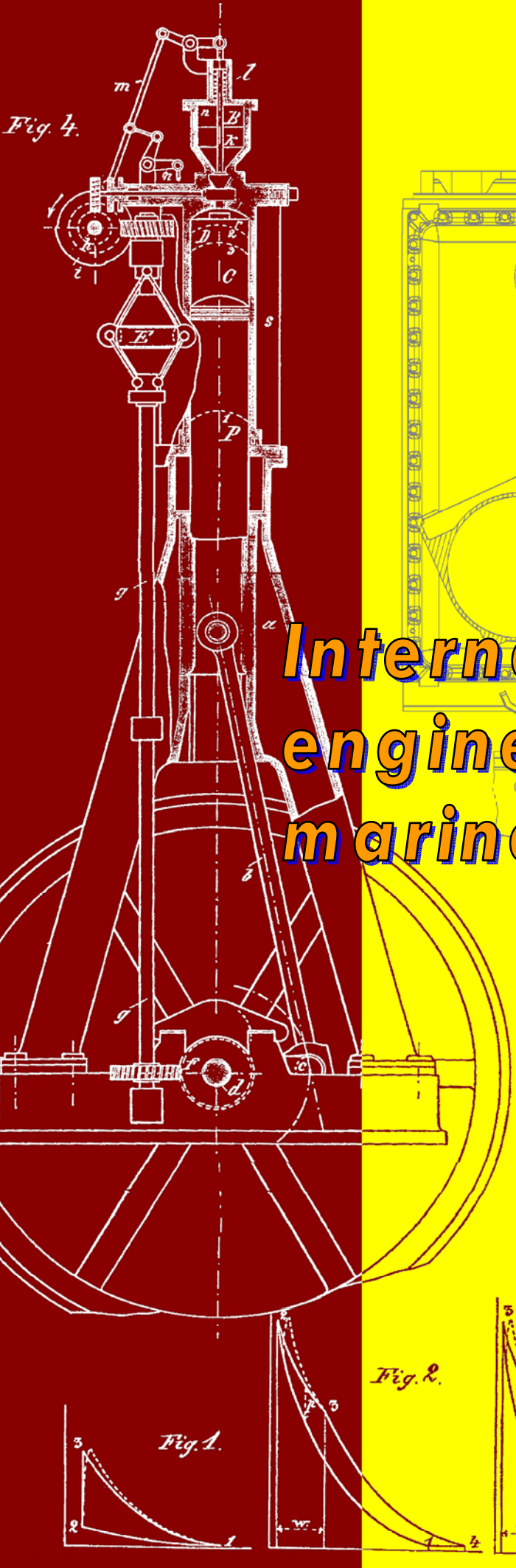
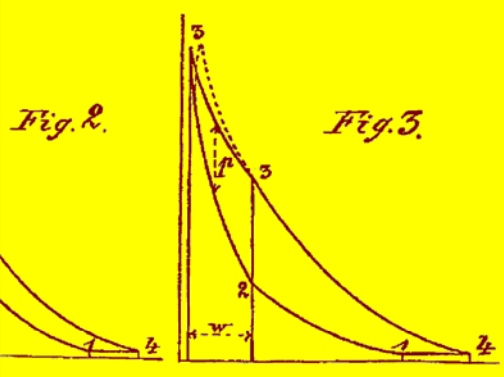


Bilousov Ievgen
Bulgakov Mykola

Internal combustion engines of modern marine vessels



https://sites.google.com/view/modern-marine-engines/

Bilousov Ievgen, Bulgakov Mykola

Internal combustion engines of modern marine vessels

Textbook for higher maritime educational institutions

Odesa – 2024

УДК 629.5.063.6/.7:629.5.03-843.6(075.8)
Б43

Approved for publication by the academic council of the
Odessa National Maritime University
(protocol No. 12, dated: July 26, 2024)
Odesa 2024.

R e v i e w e r s :
Marchenko A.P. – Doctor of Technical Sciences, Professor (National Technical University Kharkov Polytechnic Institute);
Varbanets R.A. – Doctor of Technical Sciences, Professor (Odessa National Maritime University);
Timoshevsky B.G. – Doctor of Technical Sciences, Professor (National University of Shipbuilding named after Admiral Makarov, Nikolaev).

Bilousov I.V., Bulgakov M.P.
Б43 Internal combustion engines of modern marine vessels. Textbook for higher maritime educational institutions / I.V. Bilousov, M.P. Bulgakov – Odesa, 2024, 512 p.
ISBN 978-617-7797-54-7

Over the past 20-25 years, marine internal combustion engines have undergone significant changes. These changes are reflected in this textbook. At the same time, questions regarding the design and operation features of discontinued engines are excluded from consideration or minimized. Attention is paid to the issues of increasing the economic and environmental performance of marine engines by improving pressurization systems, fuel supply, the use of gas fuel, as well as by supplying water to the engine working space. The theoretical aspects of the processes occurring in modern engines of all types and sizes are considered.

This book will be useful to anyone who studies marine internal combustion engines or is involved in their design and operation.

УДК 629.5.063.6/.7:629.5.03-843.6(075.8)

ISBN 978-617-7797-54-7 © Bilousov I.V., Bulgakov M.P., 2024

https://sites.google.com/view/modern-marine-engines/

Євген Білоусов, Микола Булгаков

Двигуни внутрішнього згоряння сучасних морських суден

Підручник для вищих морських навчальних закладів

Одеса – 2024

УДК 629.5.063.6/.7:629.5.03-843.6(075.8)
Б43

Схвалено до друку вченою радою
Одеського національного морського університету
(протокол № 12, від 26 липня 2024 р.)
Одеса 2024.

Р е ц е н з е н т и:
Марченко А.П. – доктор технічних наук, професор (Національний технічний університет «Харківський політехнічний інститут»);
Варбанець Р.А. – доктор технічних наук, професор (Одеський національний морський університет);
Тимошевський Б.Г. – доктор технічних наук, професор (Національний університет кораблебудування імені адмірала Макарова, м. Миколаїв).

Білоусов Є.В., Булгаков М.П.
Б43 Двигуни внутрішнього згоряння сучасних морських суден: підручник для вищих морських навчальних закладів / Є.В. Білоусов, М.П. Булгаков – Одеса, 2024, 512 с.
ISBN 978-617-7797-54-7

За останні 20-25 років суднові двигуни внутрішнього згоряння зазнали значних змін. Ці зміни відображені в цьому підручнику. При цьому питання щодо конструкції та особливостей роботи знятих з виробництва двигунів виключені з розгляду або зведені до мінімуму. Приділено увагу питанням підвищення економічних та екологічних показників суднових двигунів шляхом удосконалення систем наддуву, паливопостачання, використання газового палива, а також шляхом подачі води в робочий простір двигуна. Розглянуто теоретичні аспекти процесів, що відбуваються в сучасних двигунах усіх типів і розмірів.

Книга буде корисна всім, хто вивчає суднові двигуни внутрішнього згоряння або займається їх проектуванням і експлуатацією.

УДК 629.5.063.6/.7:629.5.03-843.6(075.8)

ISBN 978-617-7797-54-7 © Білоусов Є.В., Булгаков М.П., 2024

CONTENTS

SECTION 1
History of piston internal combustion engines 7
Literature used when writing section 1 11
SECTION 2.
General information on marine internal engines of combustion engines and their operation 12
2.1 General structure and principle of operation of marine piston internal combustion engines 12
2.2 Basic concepts and definitions 16
2.3 Ideal cycles of reciprocating engines 20
2.4 Real cycles of marine diesel engines 25
2.4.1 Operating procedures for a four-stroke diesel engine 25
2.4.2 Operating procedures and real cycle of a marine two-stroke diesel engine 31
2.5 Main engine performance indicators 33
2.5.1 Indicator indicators 33
2.5.2 Effective indicators 36
2.6 New approaches in organisation of operating processes in marine four-stroke engines. Miller cycle 38
2.7 Classification of marine internal combustion engines 46
2.8 Architecture of marine internal combustion engines 48
Literature used when writing section 2 53
CHAPTER 3
The design of marine internal combustion engines, the mechanisms and systems included in them. Forces and moments acting in marine engine 54
3.1 Fixed structural elements of marine engines 54
3.1.1 Foundation frame 55
3.1.2 Crankcase or crankcase box 57
3.1.3 Working cylinder 59
3.1.4 Anchor links 62
3.1.5 Cylinder covers 64
3.2 Movable structural elements of marine engines 66
3.2.1 Piston 66
3.2.2 Piston pin 71
3.2.3 Piston rings 73
3.2.4 Piston rod and rod packing 76
3.2.5 Crosshead 78
3.2.6 Connecting rods 79
3.2.7 Crankshaft group 84
3.3 Crankshaft bearings 89
3.3.1 Frame (main) bearings of the crankshaft 89
3.3.2 Crankshaft thrust bearings 91
3.4 Kinematics, dynamics of the crank mechanism and balancing of marine engines 92
3.4.1 Kinematics of the crank mechanism 92
3.4.2 Dynamics crank mechanism 97
3.4.3 Total torque from all engine cylinders 102
3.4.4 Uneven rotation of the crankshaft 103
3.4.5 External imbalance and balancing methods for marine engines 104
3.4.6 Torsional and axial vibrations 111
Literature used in writing section 3 117
SECTION 4
Air-gas systems of marine engines 119

For preliminary review

4.1 Use of atmospheric air in the operating process of marine engines 119
4.1.1 Atmospheric air 119
4.1.2 Formation of the charge of the working cylinder and parameters of its state at the moment of compression. 120
4.2 Composition of combustion products and exhaust gas parameters 124
4.2.1 The composition of the gas mixture 124
4.2.2 Harmful components in combustion products 127
4.3 Gas-dynamic processes during gas exchange 128
4.4 Purpose and main functions of air-gas systems 132
4.5 Gas distribution mechanisms of marine engines 133
4.5.1 History of the creation and development of gas distribution mechanisms for marinepiston internal combustion engines 133
4.5.2 Types of gas distribution mechanisms used 139
4.5.3 Spool valve timing 139
4.5.4 Valve timing 141
4.5.5 Combined valve-spool valve timing 142
4.5.6 Assemblies and elements of valve and valve-spool gas distribution mechanisms 144
4.5.7 Hydrostatic transmission mechanisms 157
4.5.8 Hydrostatic valve drive with electronic control 160
4.5.9 Changing valve timing when reversing the engine 166
4.5.10 Changing valve timing to optimize engine operating parameters 168
4.6 Elements of the air-gas system for receiving and purifying air 172
4.7 Supercharging units 174
4.7.1 History of the creation and development of marine diesel supercharging units 174
4.7.2 Methods of supercharging marine engines 176
4.7.3 Gas turbine charging 177
4.7.4 Turbocharger units for supercharging marine diesel engines 180
4.7.4.1 General structure and principle of operation of a radial-axial centrifugal compressor. 180
4.5.4.2 The general structure and principle of operation of the axial gas turbine stage 182
4.7.4.3 General structure and principle of operation of the radial-axial gas turbine stage 184
4.7.4.4 Design features of individual elements of turbocharger units 185
4.7.4.5 Design of rotor impellers and inlet guide vane devices 187
4.7.4.6 Design of bearings and bearing units 190
4.7.4.7 Design features of stator elements 193
4.8 Characteristics of compressors of turbocharger supercharging units 194
4.9 Adjustment of charging units 200
4.10 Engine intake tract elements 209
4.11 System elements for exhaust and noise reduction of exhaust gases 213
4.12 System elements for reducing exhaust gas toxicity 215
4.13 System elements for exhaust heat recovery 217
Literature used in writing section 4 217
SECTION 5
Fuel systems of marine diesel engines. Mixing and combustion in diesel engines, indicators operation of fuel equipment 220
5.1 Marine fuels, their composition and physical and chemical properties 220
5.1.1 Impurities in fuel 222
5.1.2 Performance characteristics of fuels and methods for their control 223
5.1.3 Marine fuel grades 224
5.2 Alternative marine fuels 226
5.2.1 Marine gas fuels 226
5.2.2 Other alternative marine fuels 227

For preliminary review

5.3 History of the creation and development of marine diesel fuel systems 229
5.4 Mechanical direct injection systems 232
5.4.1 General information about direct injection systems 232
5.4.2 Marine diesel fuel injector 235
5.4.3 Spool-controlled fuel pumps 247
5.4.4 Spool-type fuel pumps with supply advance angle control 252
5.4.5 Pump-injectors with spool control 259
5.5 Fuel pumps and valve-controlled pump-injectors 260
5.5.1 Fuel pumps with mechanically driven valves 260
5.5.2 Fuel pumps and pump-injectors with solenoid valve actuators 265
5.6 Fuel systems with hydraulically driven fuel pumps 271
5.6.1 Fuel pumps for low-speed diesel engines with hydraulic drive 271
5.6.2 Hydraulically driven pump-injectors for MAN low-speed engines 276
5.6.3 Pump-injectors with a hydraulic plunger drive and electromagnetic control of the fuel supply of high-speed Caterpillar engines 279
5.7 Accumulator injection systems 288
5.7.1 Accumulator injection systems for low-speed diesel engines of the RT-flex series from Wärtsilä 289
5.7.2 Accumulator injection systems of low-speed diesel engines with direct fuel supply control 300
5.7.3 Accumulator injection systems for medium-speed diesel engines 305
5.7.3.1 The MAN accumulator fuel supply system 306
5.7.3.2 The Wärtsilä accumulator fuel supply system 314
5.7.3.3 The Caterpillar accumulator fuel supply system 326
5.7.4 Accumulator injection systems for high-speed marine diesel engines 328
5.8 Fuel equipment of gas and gas-diesel marine engines 349
5.8.1 Converting diesel engines to engines with external mixture formation and spark ignition 349
5.8.2 Fuel equipment of four-stroke gas-diesel engines with external mixture formation 354
5.8.3 Fuel equipment of two-stroke gas-diesel engines 359
5.8.3.1 Fuel equipment of two-stroke gas-diesel engines with gas supply on the compression stroke. 360
5.8.3.2 Fuel equipment of two-stroke gas-diesel engines with gas fuel supply at the end of the compression stroke. 366
5.8.3.3 Comparison of low and high pressure systems of gas-diesel low-speed marine diesel engines. 372
5.8.3.4 Fuel system of engines running on liquefied petroleum gas and other alternative fuels. 375
5.9 Mixture formation in diesel engines. Performance indicators of the fuel equipment of modern diesel engines and their impact on the operating process 377
5.9.1 Fuel injection and atomization processes 377
5.9.2 Stages and macroscopic parameters of the injection process, their influence on mixture formation in diesel engines 377
5.9.3 Methods of mixture formation in marine diesel engines 383
5.9.4 Microscopic parameters of the injection process, their influence on mixture formation and combustion in diesel engines 388
5.9.5 Hydrodynamic processes in the fuel supply system 395
5.9.6 Injection pressure characteristic 398
5.9.7 Fuel injection characteristics 402
5.9.8 Fuel combustion in diesel engines 403
5.9.9 The influence of fuel supply characteristics on the diesel operating process 412
5.9.10 Current trends in the organization of engine operating processe 416

For preliminary review

5.10 Use of water to improve the environmental performance of marine diesel engines and equipment for supplying water to the engine 424
5.10.1 Methods for saturating incoming air with water vapor 424
5.10.2 Methods of direct water injection into the engine working space 426
Literature used when writing section 5 429
SECTION 6
Marine engine oil systems 435
6.1 Purpose and composition of the system 435
6.2 Circulating oil systems 436
6.3 Cylinder lubrication systems 445
6.4 Turbocharger lubrication systems 455
6.5 Friction and lubrication modes 457
Literature used when writing section 6 459
SECTION 7
Heat flows in marine engines. Heat removal systems from their structural elements 461
7.1 Thermal balance of marine engines 461
7.2 Heat exchange between the engine working space and the cooling medium 463
7.3 Temperature fields of parts of the cylinder-piston group 465
7.4 Thermal stresses of parts of the cylinder-piston group and factors limiting the temperature of the working surfaces 467
7.5 Purpose and composition of cooling systems 470
7.5.1 High temperature circuit 472
7.5.2 Piston cooling 473
7.6 Charge air cooling 477
7.7 Main units and elements of cooling systems 477
Literature used when writing section 7 483
SECTION 8
Starting and reversing systems 484
8.1 Engine starting process 484
8.2 Electric starting 484
8.3 Pneumatic systems for starting and reversing engines 485
8.3.1 Pneumatic starters 485
8.3.1.1 Pneumatic starters with positive displacement engines 485
8.3.1.2 Pneumatic starters with dynamic engines 487
8.4 Direct starting systems 489
8.5 Electronically controlled direct starting systems 494
Literature used when writing section 8 497
Annex 1. List of abbreviations 499
Annex 2. Brief information about scientists and engineers who made a significant contribution to the development and establishment of engine building 501

For preliminary review

Introduction

Marine internal combustion engines have firmly taken leading positions in the global merchant fleet as the main and auxiliary energy sources that provide both the propulsion of the vessel and the generation of electrical and other types of energy used to support technological processes, the life of the crew and passengers. Due to the peculiarities of the organization of the operating process, in which fuel combustion occurs directly in the working cylinders, internal combustion engines today have the highest efficiency in converting thermal energy into mechanical work. In addition, the use of heavy fuels obtained as residual products of oil refining in marine diesel engines has significantly reduced the cost of generated energy. The introduction of new technologies related to the use of electronic control systems, as well as the transition to gas and gas-diesel cycles, has made it possible to successfully address issues of increasing the environmental performance of ship engines. Internal combustion engines are relatively simple and technologically advanced to operate, and have a long service life, often commensurate with the service life of the vessel on which they are installed.

All of the above has led to the fact that today 98% of the world's merchant fleet ships are equipped with internal combustion engines. Marine engine building is a leading sector of the world economy, which is dynamically developing and strives to respond to market demands as quickly as possible.

Currently, marine internal combustion engines are produced by several dozen manufacturers who have their own approaches to solving certain problems related to increasing fuel efficiency, environmental friendliness, reliability and reducing operating costs. Generalization of the accumulated experience in solving these problems can be useful to specialists engaged in both the design and operation of engines, as well as undergraduate and graduate students studying in maritime educational institutions. It is for this audience that the book offered to your attention is intended. The information provided in the book is accompanied by drawings, drawings and other graphic material that give an idea of the design solutions used by one or another manufacturer.

It should be noted that the volume and structure of information provided by different manufacturers may vary greatly, so the data presented in the book may differ in the completeness of presentation. The information provided in the book can only be used for preliminary acquaintance with the main design solutions and theoretical foundations of the processes occurring in engines. To obtain more detailed information, please refer to specialized literature.

The information provided in the book maximally corresponds to the requirements for the training of ship mechanics, both at the operational and management levels, set out in the International Convention on Standards of Training, Certification and Watchkeeping for Seafarers, 1978.

Based on this, the main purpose of the book is its use in the educational process of maritime educational institutions that train maritime specialists for the merchant fleet. At the same time, the book will be useful to existing mechanics, as well as to everyone involved in the design and operation of marine piston internal combustion engines.

The authors express special gratitude to the reviewers for a number of valuable comments made during the preparation of the manuscript. The authors also express gratitude to the teams of the Department of Operation of Ship Power Plants of the Kherson State Maritime Academy and the Department of Ship Power Plants and Technical Operation of the Odessa National Maritime University for their assistance in collecting and preparing materials.

For preliminary review

SECTION 1

History of piston internal combustion engines

The history of reciprocating internal combustion engines (ICEs) dates back to 1678. In this year, the first piston engine was proposed by the outstanding Dutch mathematician, mechanical engineer and physicist Christian Huygens (1629-1695). It was assumed that black powder would be used as fuel for such an engine. However, Huygens' proposed engine was never built.

Proposed in 1860 by the Belgian engineer Etienne Lenoir* (1822-1900), the engine worked without compression and its efficiency was slightly different from steam engines, so it did not compete significantly with them. However, given its relative simplicity due to the absence of a steam boiler, this engine became quite widespread.

The idea of organising the operating process with pre-compression of the working medium, expressed as early as 1828 by Sadi Carnot*, was later implemented by Eugene Langer* (1833-1895) and Nicolos Otto* (1832-1891) for the four-stroke engine they designed. This engine consumed four times less fuel than Lenoir's engine and was the first efficient reciprocating internal combustion engine.

The first internal combustion engines ran on generator gas, which created certain inconveniences for their use, especially in transport. As a result of numerous searches for universal fuels for internal combustion engines, the choice fell on petroleum products. This type of fuel was further associated with a significant improvement of Otto engines and their widespread use in transport, but the efficiency of such engines remained quite low for a long time.

Attempts to create a piston internal combustion engine with high fuel efficiency were made by many engineers of that time, but the greatest success was achieved by the German engineer Rudolf Diesel* (1858-1913) (Fig. 1.1). On 28 February 1892, he was granted patent No. 67207 for «Operating process and Method of Execution of a Single-Cylinder and Multi-cylinder Engine». About a year after the patent was granted, Diesel's brochure entitled «Theory and Construction of a Rational Heat Engine to Replace the Steam Engine and Other Presently Existing Engines» was printed by the technical literature publisher Julius Springer in Berlin. In this brochure, Diesel, analysing the experience of creating piston engines, as well as the basic theoretical provisions of thermodynamics, including those developed by the founder of technical thermodynamics Sadi Carnot* in his work «On the motive power of fire and on machines capable of transforming this power», proposed the concept of creating a rational heat engine.

It is interesting to note that Diesel originally intended to use fine coal dust as fuel for his engine, which was to be fed into the cylinder at the end of the compression process by a jet of compressed air. Diesel was convinced that the energy of the air charge, as a result of high compression, could exceed the activation threshold of the fuel combustion process and, as a consequence, self-ignition would occur in the engine cylinder. For this purpose, it was proposed to significantly increase the compression ratio of the air charge compared to the Otto engine.

Diesel's work aroused great interest among specialists, finding both supporters and opponents. In fact, a fundamentally new type of engine with gradual combustion of fuel atomised in an air charge was proposed. Many assessed the provisions outlined in the brochure as extremely interesting in theoretical terms, but hardly feasible in practice. Nevertheless, there was no doubt in anyone's mind that, if the theoretical premises described by Diesel could be realised in practice, the engine thus obtained would have unquestionable advantages over all heat engines known at the time.

* brief information about this figure of science and technology is given at the end of the book

For preliminary review

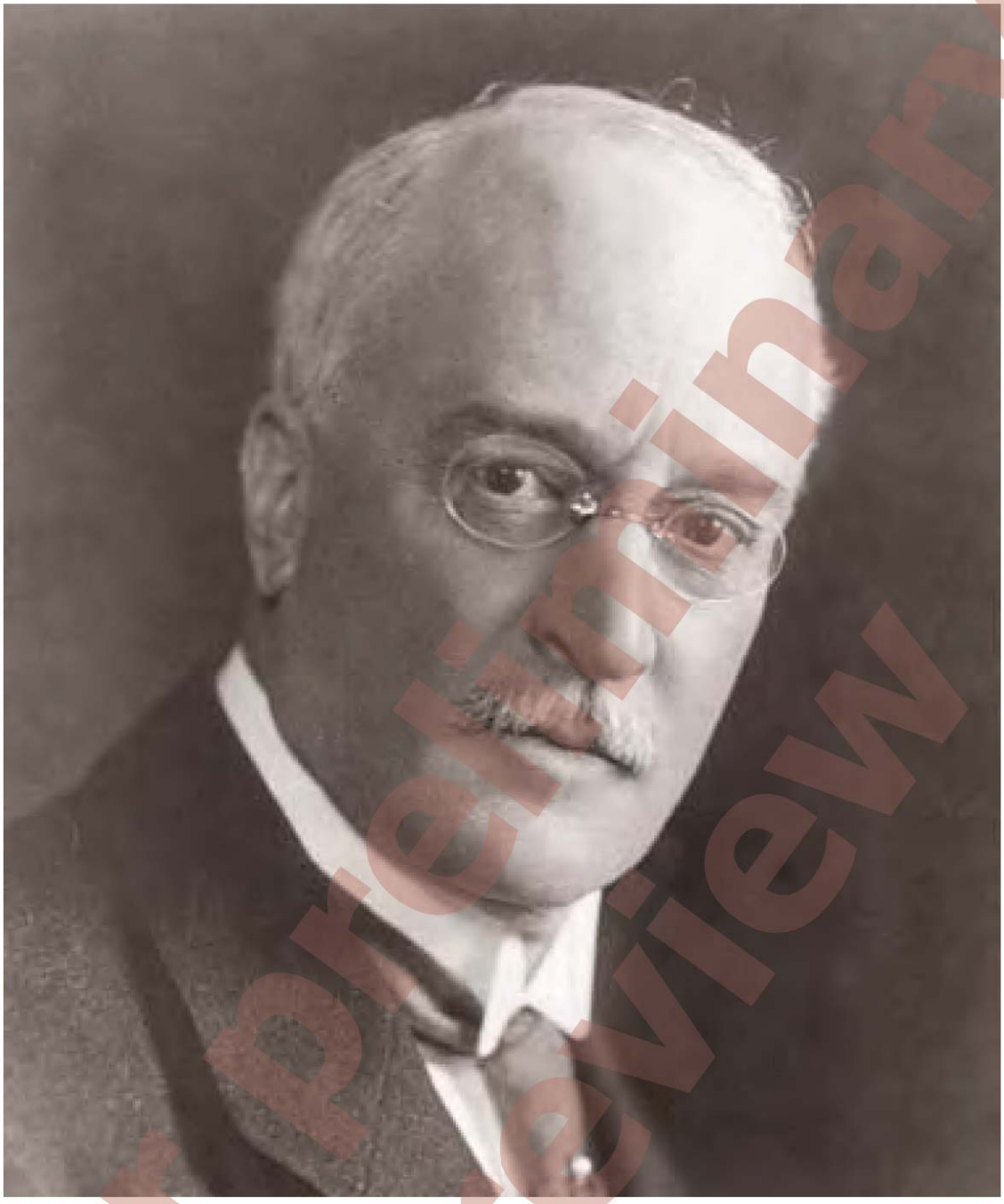


Figure 1.1 – Rudolf Diesel [1]

In February 1893, the Augsburg Machine Works (now the MAN Group), represented by its director G. Buza*, agreed to provide funds for the construction of an experimental engine. Later, in April of the same year, Friedrich Krupp's steelmaking company in Essen joined in financing the project.

Successively from 1893 to 1897 R. Diesel produced three prototypes. The first one fully complied with the principles outlined in his patent, however, faced with difficulties in feeding coal dust into the combustion chamber, the inventor decided to first test the engine on liquid fuel – lamp kerosene. The third engine (Fig. 1.2) built by Diesel, running on kerosene, showed an efficiency of about 26.2%, which was one and a half to two times higher than the efficiency of other heat engines known at that time.

For preliminary review

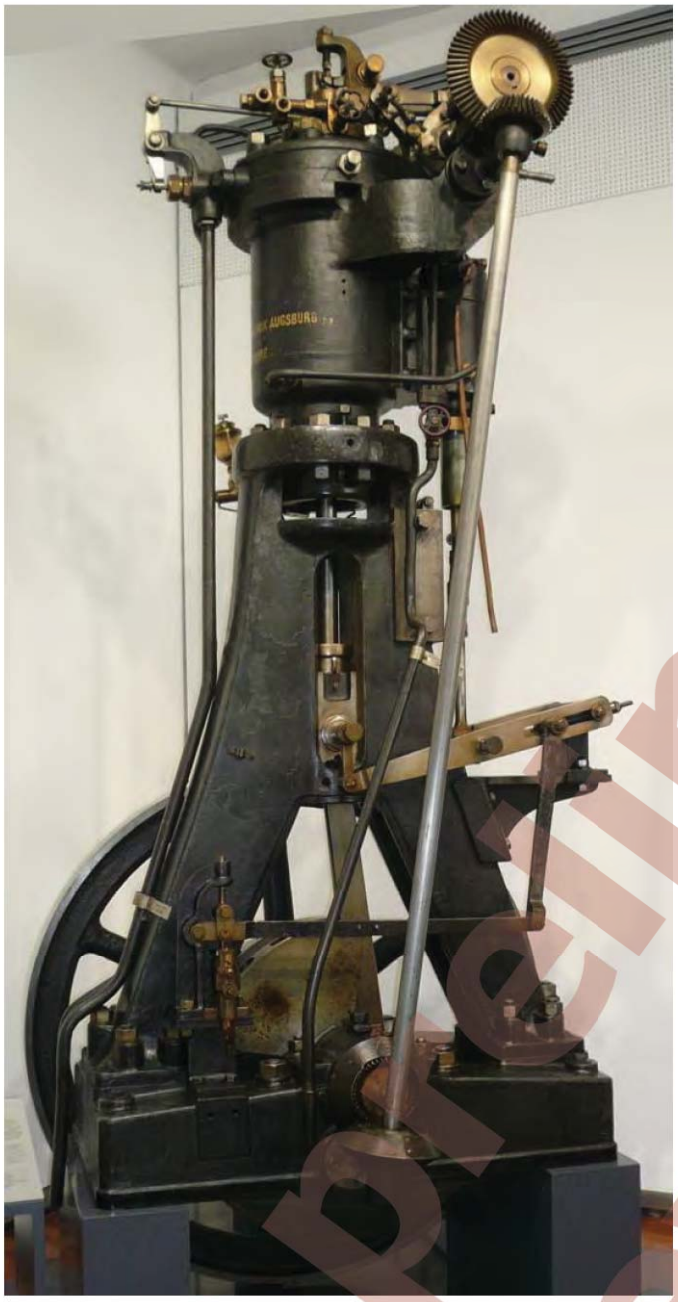


Figure 1.2 – One of the first workable R. Diesel engines [1]

Therefore, even in this form, the Diesel engine began to enjoy commercial demand, and the main efforts of the inventor, as well as most other engineers, were directed at improving the operating process based on the use of liquid fuels.

Among those who recognised the prospects of the new invention for use as a marine engine was Joan Jacob Sulzer* (1855-1922), a descendant of the founders of the Sulzer Brothers* company, which at that time was manufacturing marine steam engines in Winterthur, Switzerland. In 1879, R. Diesel, while still a student at the Technical High School in Munich, had an internship at this company, and with the son of the founder of the company he was connected not only by business relations, but also by personal friendship. In 1898 the Sulzer brothers' firm acquired the engine patent and during the next few years completely reoriented its production to the production of new products. One of the first marine diesel engines produced at this company is shown in Fig. 1.3. In 1905, the Sulzer brothers' factory was the first to design two-stroke engines, which are now predominantly used in the merchant marine.

The owners of the Danish engineering concern Burmeister & Wain* had a similar foresight. They acquired a patent in 1898 and started production of stationary and marine Diesel engines. The first engine of this concern has survived to the present day (Fig. 1.4).

For preliminary review

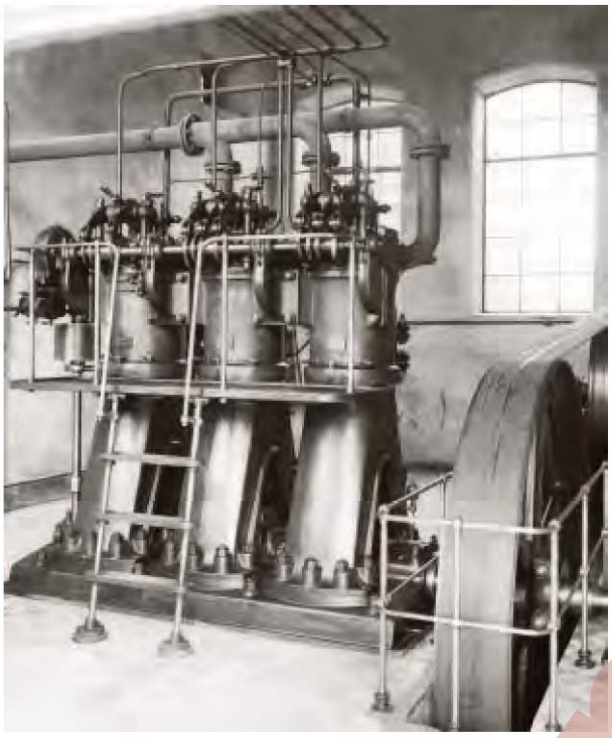


Figure 1.3 – The first 120 hp Sulzer diesel engine built for the Walenstadt cement factory in 1904 [1].

At first, the production was carried out according to the original drawings given by R. Diesel, but later a number of design flaws were identified, and the concern's engineers developed their own design of the engine taking into account the peculiarities of operation in shipboard conditions.

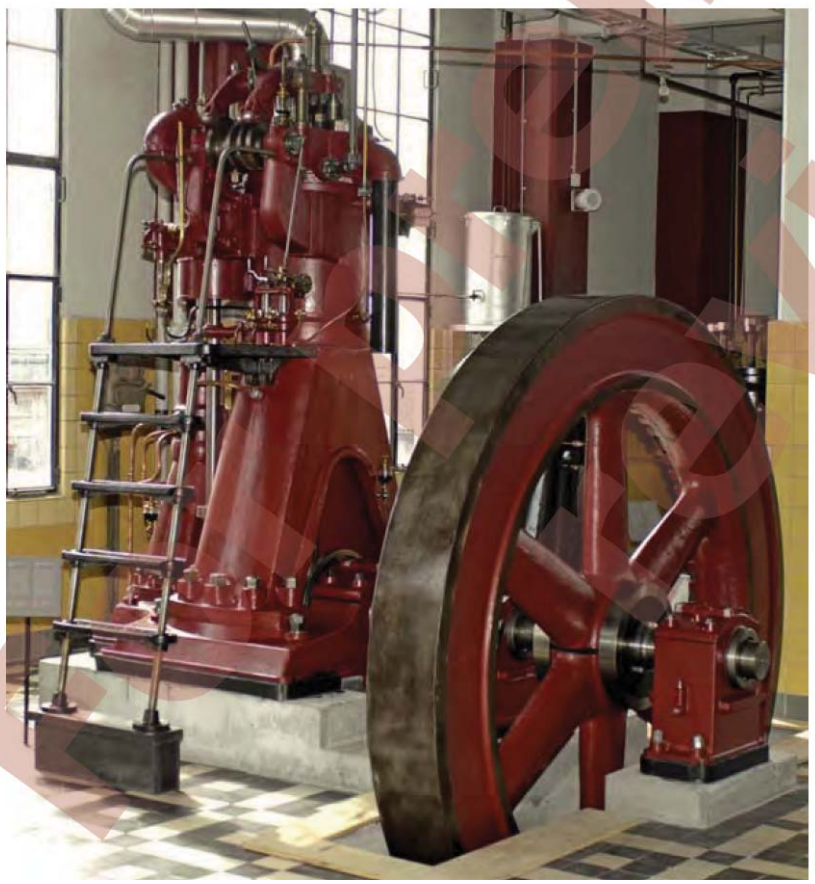


Figure 1.4 – Burmeister & Wain Engine No. 1

Figure 1.4 shows the model DM140 B&W engine, which has a cylinder bore of 320 mm and a piston stroke of 490 mm. The engine has a power output of 40 horsepower at 180 min^{-1} and weighs 16 tonnes. The engine ran reliably for 40 years and in 1940, in perfect working order, was bought by B&W for the Christianshavn Museum in Copenhagen. The engine has now been fully restored and is in working order.

Preliminary view

In 1912, at the shipyard of Burmeister & Wain* near Copenhagen, the first ocean-going engine ship «Selandia» is built with two eight-cylinder four-stroke reversible marine diesel engines with an output of 932 kW, at 140 min^{-1}. From this moment on, the diesel engine begins to confidently win back its position in the world merchant fleet.

The new engine finds supporters in the Russian Empire. One of the first to acquire Diesel's patent was Emmanuel Nobel*, who owned a number of engineering plants in Russia, particularly in St. Petersburg.

On the recommendation of his friend, Professor Georgiy Depp* of the St. Petersburg Institute of Technology, Nobel bought Diesel's patent in 1898. As early as 1899, not only was the production of diesel engines established, but for the first time a diesel engine was installed on the river vessel Vandal, reversible engines and engines fuelled by unrefined oil were developed.

Russian scientist G.V. Trinkler*, in parallel with French L.G. Sabathé* developed and proposed a diesel engine cycle, with fuel ignition from compression with mixed heat supply first at constant volume and then at constant pressure.

All modern compression-ignition internal combustion engines operate on the Trinkler-Sabathé cycle.

In 1906, Professor V. Grinevetsky* of the Higher Moscow Technical School developed the theory of the operating process of internal combustion engines, which was further deepened by his students – Professors E. Mazing* and N.R. Briling*. The theory is characterised by simplicity of mathematical apparatus, clarity of definitions, correctness of qualitative assessment of influence of various factors on the whole operating process. This theory has long served as a basic theory in the study of operating processes in the engine cylinder, and is still an important aid in the educational design of internal combustion engines; it provides a successful choice of the main parameters that give a visual representation of the peculiarities of the course of individual processes. This theory is an excellent tool for the clarification of basic relationships and is indispensable for pedagogical purposes.

The theory of gas exchange in two-stroke engines was developed by Prof. A. Orlin. A great contribution to the theory and practice of marine diesel engines was made by V. Vansheidt* who for many years headed the team of designers of the plant «Russian Diesel».

Professor N.M. Glagolev*, who for a long time headed the department «Internal Combustion Engines» at the Kharkiv Polytechnic Institute, developed a method of modelling the engine operating process on the basis of mass balance. Professor N. Razleitsev* developed a method for calculating the process of atomisation and combustion of fuel.

Unfortunately, it is impossible to list all those whose names are deservedly inscribed in the history of world diesel engineering in this book. At present the work on improvement of marine engines is continued by a pleiad of young scientists, each of whom contributes to the further development of the industry with his labour.

Some of them are only at the beginning of their way to the top of scientific and engineering activity in the field of design and operation of marine engines. Perhaps this book will be their first step on this path, and who knows, maybe in the future new authors will include their names in their works on the history of the industry.

Literature used when writing section 1:

1. Rudolf Diesel (1858-1913). First Marketable Diesel Engine (1897). MAN Diesel SE. 2006 – 76 p.

2. MAN Group History 1758-2006. MAN AG. 2006 – 29 p.

3. Kontinuität und Wandel in 175 Jahren Sulzer. Technische Meilensteine gesetzt. Sulzer Technical Review. 2009 – P. 31-34

4. B&W No. 1 Comes Back to Life. The historic MAN Diesel engine and its successful restoration in DieselHouse. DieselFacts. MAN Diesel A/S. Copenhagen, Augsburg. 2007 – P 12.

For preliminary review

SECTION 2.

General information on marine internal engines of combustion engines and their operation

2.1 General structure and principle of operation of marine piston internal combustion engines

Diesel internal combustion engines (hereinafter referred to as diesels) are predominantly used in the fleet. They are installed on 95...98% of sea and river vessels. Other types of engines are rarely used on merchant ships, and their consideration is beyond the scope of this course. Therefore, hereinafter the term internal combustion engine (ICE) or marine internal combustion engine (MICE) will be used to refer to a piston engine of the diesel type.

An internal combustion engine is a reciprocating heat machine in which the chemical energy of the fuel is first converted into heat, and then the heat energy is converted into mechanical work.

All processes related to the conversion of fuel energy into mechanical work in this class of heat engines take place in the space of the working cylinder. Heat is supplied to the working medium in the process of fuel combustion in the combustion chamber. At the first stage the working medium is air (filling and compression processes), and at the second stage (after ignition and combustion of fuel) – gaseous products of fuel combustion, which, expanding, do useful work.

This method of energy conversion has significant advantages over other types of heat machines, as the hot heat source is located inside the engine itself. As a result, there are no losses associated with heat exchange, as is the case, for example, in the cycles of steam power plants. As a result, the engine is compact, as there are no bulky and inefficient heat exchangers.

The rapidity of the operating process makes it possible to trigger higher temperature differences in the internal combustion engine without damaging the structural elements of the engine itself. This in turn leads to an increase in the thermal and effective efficiency of the cycle. Today marine diesel engines are the most efficient heat engine available to mankind. The best samples of such engines have an efficiency of 48...54%, and with heat recovery systems this figure is even higher.

All modern internal combustion engines are designed to force high-pressure air into the engine cylinders. This method of air supply is called supercharging. The use of supercharging provides higher mass filling of the cylinders, which makes it possible to increase the cycle fuel supply (compared to un-supercharged internal combustion engines) and, consequently, to obtain greater effective power at the engine crankshaft output flange.

The most common is gas turbine supercharging, where the energy released by the expansion of exhaust gases on the blades of the gas turbine is used to drive a centrifugal compressor, which precompresses the air to the required parameters. Such supercharging units are called gas turbochargers.

Studies of the operating process have shown that this method of supercharging allows, through the use of exhaust gas energy, to increase the efficiency of the diesel engine, its economy and the power removed from the crankshaft output flange.

In the case of two-stroke engines, the introduction of a gas turbine supercharger allowed to simplify the purge scheme considerably, eliminating the need for bulky purge pumps. The combination of a piston engine and a gas turbocharger in one power unit affects all thermal and gas-dynamic processes in the engine cylinders.

An internal combustion engine is a piston-type thermal machine in which the chemical energy of the fuel is first converted into heat, and then the heat energy is converted into mechanical work.

For preliminary review

Engines combining a piston machine, gas turbine and compressor in one unit are called combined engines. The general structure of a modern marine two-stroke engine with gas turbine supercharging is shown in Fig. 2.1.

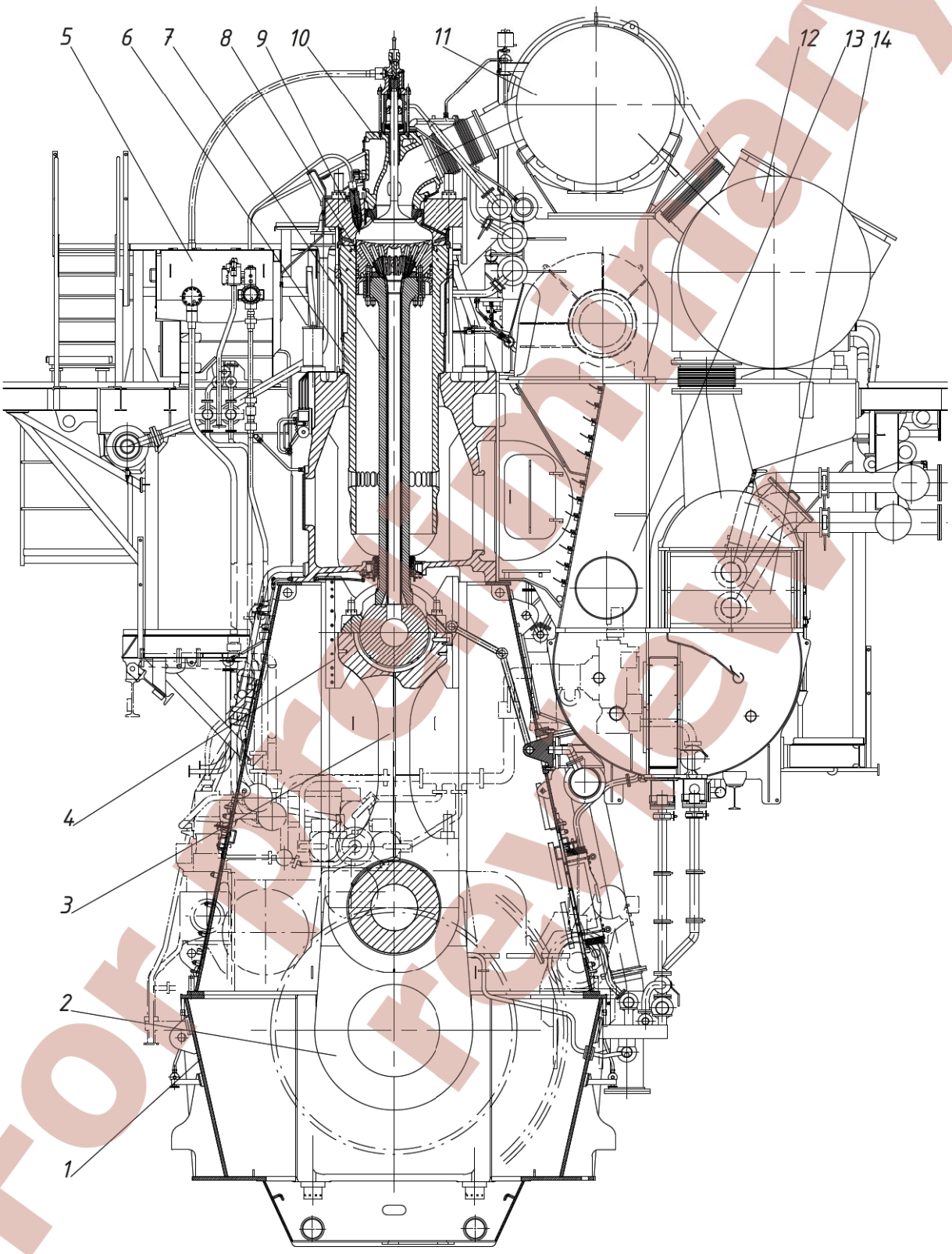


Figure 2.1 – General structure of the RT-flex 82C two-stroke crosshead diesel engine: 1 – frame; 2 – crankshaft; 3 – connecting rod; 4 – crosshead; 5 – fuel supply and gas distribution control unit; 6 – cylinder sleeve; 7 – piston rod 8 – piston; 9 – cylinder cover; 10 – exhaust valve actuator; 11 – exhaust receiver; 12 – turbocharger; 13 – intake receiver; 14 – charge air cooler (adapted from [1])

For preliminary review

Such engines form the basis of ship power in the world merchant fleet, they are installed in approximately 75% of ships of various purposes.

Along with low-speed two-stroke diesel engines, four-stroke diesel engines are widely used on ships, which are used as main engines, as well as for driving auxiliary mechanisms and units (ship power plants, compressors, etc.). The general layout of a marine four-stroke diesel engine is shown in Fig. 2.2.

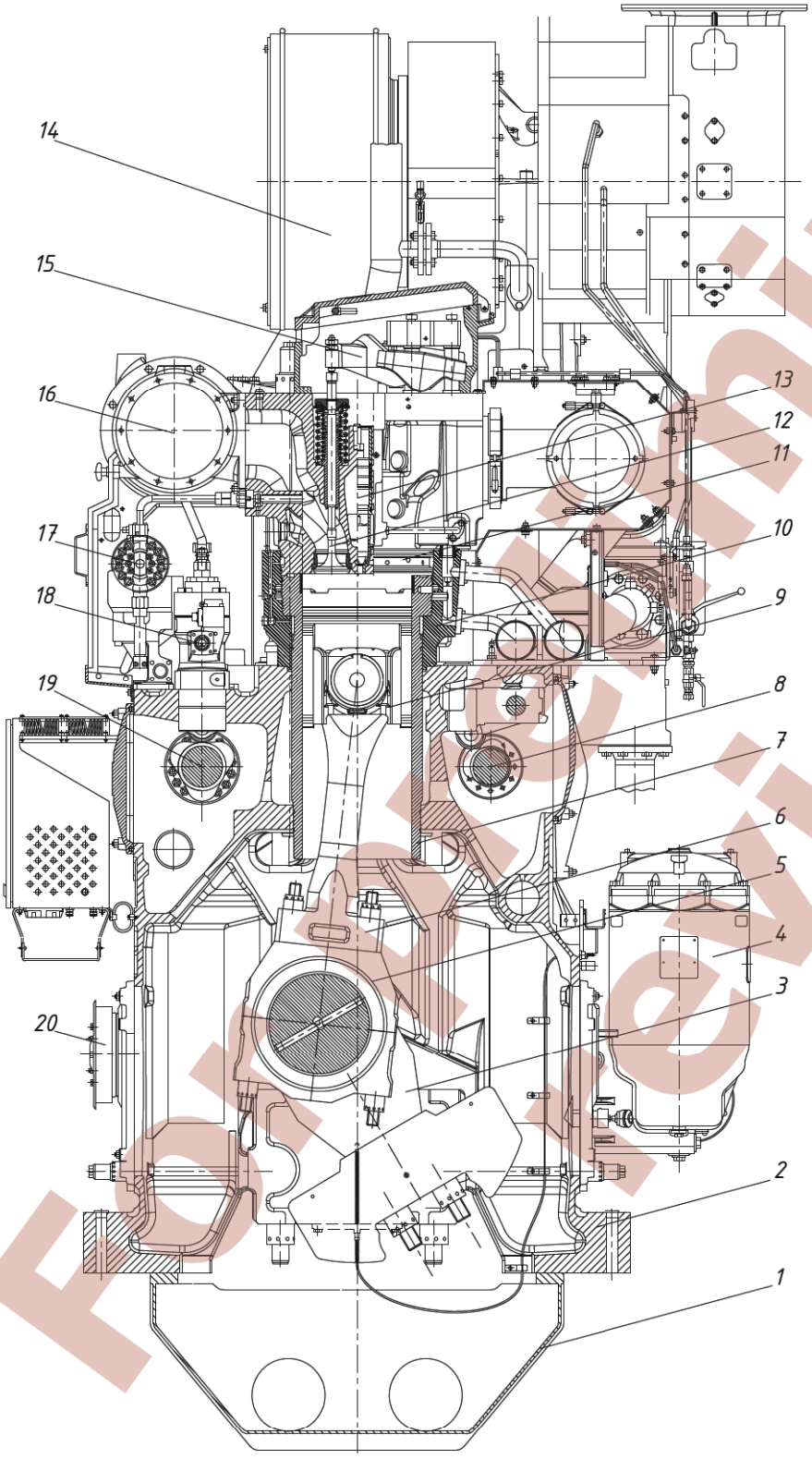


Figure 2.2 – General layout of the marine four-stroke MAN L32/44 trunk diesel engine. 1 – crankcase pan; 2 – block-crankcase; 3 – crankshaft; 4 – oil filter; 5 – lower connecting rod head bearing; 6 – connecting rod; 7 – cylinder sleeve; 8 – timing valve drive shaft; 9 – piston; 10 – cooling jacket; 11 – cylinder cover; 12 – timing valves; 13 – fuel injector; 14 – turbocharger; 15 – timing valve drive rocker arms; 16 – inlet receiver; 17 – fuel accumulator; 18 – fuel injector; 19 – fuel injector drive shaft; 20 – engine crankcase safety valve (adapted from [1])

Two-stroke high-speed engines are used less often in marine installations, however, some manufacturers give preference to this type of engine, which, all other things being equal, has a greater

For preliminary review

liter power. An example of such an engine EMD 645 from Electro-Motive Diesel is shown in Fig. 2.3.

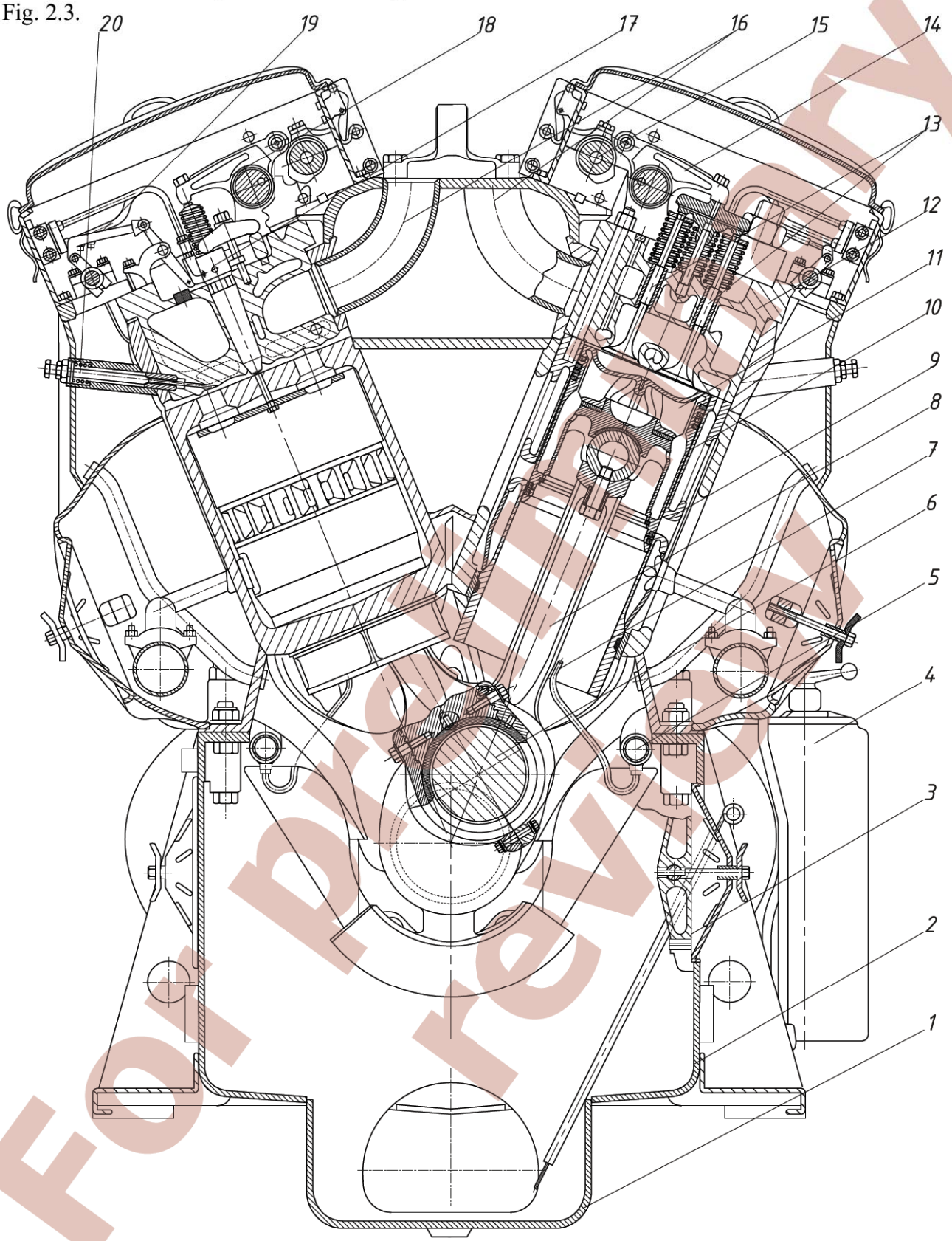


Figure 2.3 – Two-stroke high-speed V-shaped engine of EMD 645 series by Electro-Motive Diesel: 1 – crankcase sump; 2 – foundation frame; 3 – crankcase hatch; 4 – oil filter; 5 – lubricating oil line; 6 – crankshaft; 7 – oil nozzle for piston cooling; 8 – connecting rod; 9 – purge ports; 10 – cylinder sleeve; 11 – piston; 12 – cylinder cover; 13 – exhaust valves; 14 – valve drive mechanism; 15 – camshaft; 16 – exhaust manifold; 17 – injector pump; 18 – injector pump drive mechanism; 19 – fuel supply control mechanism; 20 – indicator tap (adapted from [2])

For preliminary review

Thus, all engines are divided into two-stroke and four-stroke engines according to the principle of operation. All processes in internal combustion engines are associated with the movement of the piston inside the working cylinder, which performs reciprocating motion, converted by the crankshaft into rotary motion. In two-stroke engines, the entire cycle of processes associated with obtaining useful work is carried out for one revolution of the crankshaft, in four-stroke engines for two.

For further presentation of the principles underlying the operation of reciprocating internal combustion engines, it is necessary to define the basic concepts and definitions.

2.2 Basic concepts and definitions

The general scheme of the basic geometrical relations of the working cylinder and crank mechanism of the trunk engine is shown in Fig. 2.4.

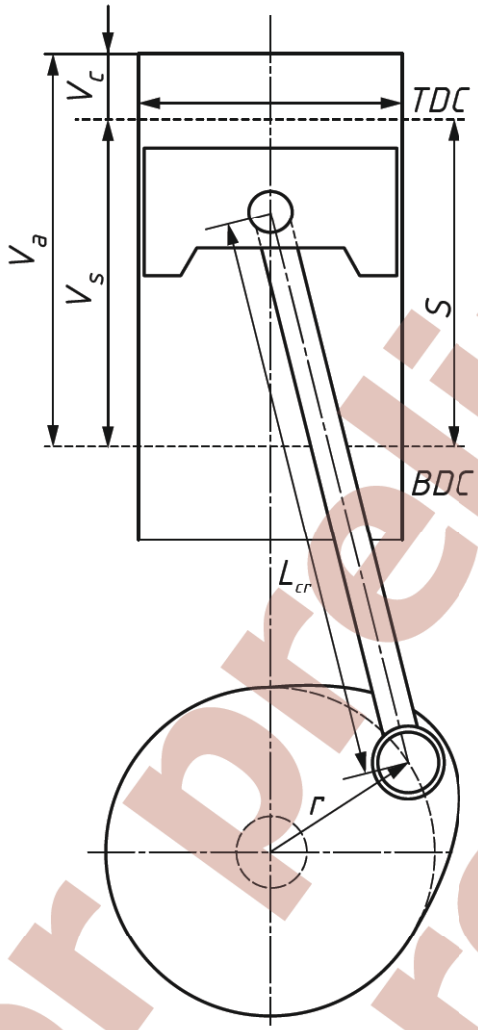


Figure 2.4 – Basic geometric relationships of the slave cylinder and crank mechanism

As already mentioned, when the shaft rotates, the piston makes a reciprocating movement in the cylinder and alternately occupies its end positions. The crank-rod mechanism (CRM) shown in Fig. 2.4, in which the cylinder axis intersects with the crank axis, is called central. This type of mechanism is typical for most marine engines. The positions of the crank-rod mechanism (Fig. 2.4), at which the axes of the connecting rod and crank lie in the same plane passing through the cylinder axis, are called *dead points*, because at these positions the force applied to the piston does not create torque on the crankshaft.

In the case of the centre mechanism, the dead centres correspond to the extreme upper or extreme lower positions of the piston in the cylinder.

Top dead centre (TDC) – is the position of the piston at which the distance from the piston to the crankshaft axis is greatest.

For preliminary review

Bottom dead centre (BDC) – is the position of the piston at which the distance from the piston to the crankshaft axis is smallest.

The cylinder diameter D – is the main geometric dimension of the engine and the choice of other engine dimensions depends on it to a large extent.

Piston stroke S is the distance travelled by the piston from one dead centre to another. In the central mechanism, each piston stroke corresponds to a 180° rotation of the crankshaft.

The *crank radius r* – is the distance between the frame and crank journals of the crankshaft. For a centre crank mechanism, the piston stroke $S = 2r$ (for each specific engine, S and r are constant values).

The volume of the compression chamber or combustion chamber V_c, is the volume of the cylinder above the piston at TDC.

The working volume of a cylinder V_s, is the volume displaced or released by the piston in one stroke from BDC to TDC and vice versa:

$$V_s = (\pi D^2 S)/4.$$

The total cylinder volume V_a, is the volume above the piston at its position at BDC.

$$V_a = V_s + V_c$$

The length of the connecting rod l_{cr}, largely determines the height of the engine, as well as the nature and magnitude of the acting forces and moments in the crank mechanism.

The theoretical or *geometric compression* ratio ε is the ratio of the total cylinder volume to the volume of the compression chamber:

$$\varepsilon = \frac{V_a}{V_c} = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c}.$$

The compression ratio is a dimensionless value that shows how many times the volume of the cylinder above the piston decreases, i.e. the charge in the cylinder is compressed, when the piston is moved from the BDC to the TDC.

The theoretical compression ratio is used for approximate calculations of operating processes of four-stroke engines, in which it can be conventionally considered that the compression process begins at BDC (except for four-stroke engines operating on the Miller cycle).

In all types of engines, with increasing compression ratio, heat utilisation improves and, consequently, the indicator efficiency of the engine increases (Fig. 2.5), which has a positive effect on their performance characteristics. However, at the same time at medium and maximum loads, the toxicity of exhaust gases increases (the amount of hydrocarbons due to the increase in the volume of the wall layer during combustion and the amount of nitrogen oxides due to the increase in combustion temperature). The load on the crank-rod mechanism also increases, and in order to ensure reliable operation of the engine it is necessary to increase the size and weight of its main parts accordingly. As a consequence, mechanical losses increase, which gradually offset the advantages associated with the increase in thermal efficiency (Fig. 2.5).

In four-stroke diesel engines, as the compression ratio increases, the temperature and pressure at the time of fuel injection start increase, resulting in a lower pressure build-up rate ($dp/d\varphi$) and softer engine operation. Increasing ε also improves the starting properties of the engine. Indicator efficiency in the range of compression ratios (ε = 14.5...19) used in diesel engines changes insignificantly. Temperature increase, at large values of ε, leads to an increase in the content of nitrogen oxides in the combustion products.

In practice, the lower limit ε, is determined by the need to achieve in the working cylinder at the end of compression temperature providing guaranteed autoignition of the fuel supplied, the upper limit – a rational ratio of thermal efficiency and mechanical losses (shaded area in Fig. 2.5).

In two-stroke engines, due to the specifics of the operating process, part of the working stroke of the piston is lost to organise gas exchange between the working cylinder and the environment. To remove combustion products and supply fresh air, the cylinder has special purge ports or ports

For preliminary review

combined with valves (Fig. 2.6). The compression process in this case starts only after all the gas distributing organs are closed.

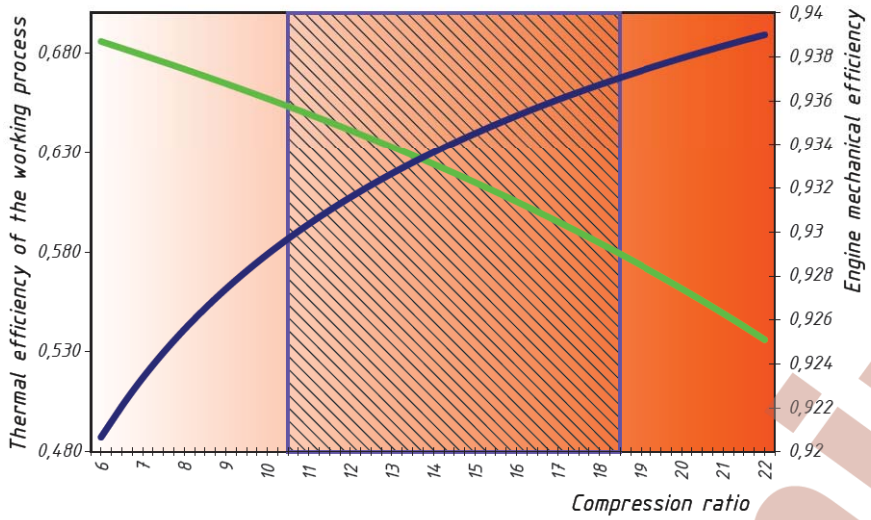


Figure 2.5 – Variation of the thermal efficiency of the operating process (η_t) and mechanical efficiency (η_{mech}) depending on the compression ratio

To take into account the losses of piston stroke for gas exchange organs, the concept of actual compression ratio is introduced.

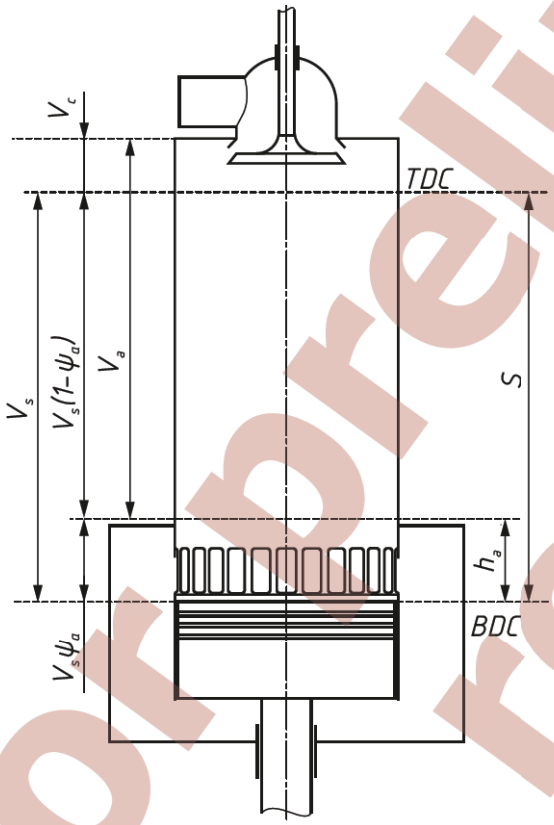


Figure 2.6 – Basic geometrical relations of the working cylinder of a two-stroke engine (h_a – part of the piston stroke at which the exhaust valve remains open)

The *actual compression* ratio ε_0 is the ratio of the cylinder volume at the beginning of compression to the combustion chamber volume:

$$\varepsilon_0 = \frac{V_c + V_s(1-\psi_a)}{V_c}$$

where $\psi = h_a/S$ – the share of the lost stroke on the gas exchange organs, after the closure of which compression begins in the cylinder; h_a – the distance travelled by the piston from the BDC to the moment of closing the exhaust valve, *m* (Fig. 2.6).

If the closing angle of the camshafts (compression start angle – φ_{csa}) is known, the value of the lost travel h_a, can be determined from the expression:

For preliminary review

$$h_a = r\left[(1-\cos\varphi_{csa})+\frac{\lambda_{cr}}{4}(1-\cos 2\varphi_{csa})\right]$$

where r – is the radius of the engine crank, m; λ_{cr} – is the constant of the crank mechanism (will be considered further on).

There is a relationship between the theoretical and actual degrees of compression:

$$\varepsilon_0 = (\varepsilon - \psi_a)/(1-\psi_a),$$

from where:

$$\varepsilon = \varepsilon_0(1-\psi_a)+\psi_a.$$

In calculations of two-stroke diesel engines, the cylinder volume at the beginning of compression refers to the cylinder volume at the moment of closing the exhaust ports or valves:

$$V_a = V_c + V_s(1-\psi_a).$$

The term «compression ratio» usually refers to the geometric compression ratio for four-stroke engines and the actual compression ratio for two-stroke engines.

The ratio of piston stroke to cylinder diameter S/D – is the main engine dimension. Not only the power but also the mass-dimensional parameters of the internal combustion engine depend on the value of this ratio to a great extent.

As *S/D* decreases:

- reduces the height of the engine and increases its length;
- mechanical losses and wear rate of parts are reduced by reducing the average piston speed;
- the valve positioning conditions are improved and the prerequisites for increasing the valve size are created;
- stiffness of crankshafts is increased due to the possibility of increasing the diameters of main and connecting rod journals and their overlapping.

It is reasonable to reduce the *S/D* ratio with increasing engine speed. Such a tendency is observed for high-speed four-stroke marine four-stroke engines, which in recent years have seen a decrease in *S/D* from 1.4 to 1...1.1 with a proportional increase in crankshaft speed.

Increasing the cylinder diameter, and the resulting decrease in *S/D* leads to:

- increase of load on CRM parts from gas pressure and inertia forces;
- reduction of the combustion chamber height and deterioration of mixing conditions.

Due to the above-mentioned reasons, there is a tendency to increase *S/D* for modern medium- and low-speed engines. Thus, for low-speed diesel engines of the latest series *S/D* has increased from 2.16...2.67 to 2.83...3.82 and even up to 4.8 due to the increase in piston stroke.

Constant of the crank mechanism λ_{cr}, is the main dimensionless design parameter of the crank mechanism, which is found as the ratio of the crank radius to the connecting rod length $\lambda_{cr} = r/L_{cr}$. By connecting rod length L_{cr} is meant the distance between the axes of its upper and lower heads.

This value directly affects the kinematics and dynamics of the engine. As λ_{cr} increases, becomes more compact, but the normal forces tending to press the piston against the cylinder liner (in a trunk engine) or the ram against the parallel (in a crosshead engine) increase.

In modern marine internal combustion engines λ_{cr} lies within the range:

- high-speed engines – 0.25...0.32;
- medium speed engines – 0.35...0.45;
- low-speed engines – 0.42...0.52.

Two-stroke and four-stroke internal combustion engines differ in the way the working cycle is organised.

The *operating cycle* is a complex of consecutive periodically repeating processes in the engine working cylinder.

A *process* is a change in the state of a working medium accompanied by a change in its parameters.

Stroke is part of the operating process that occurs in one pass of the piston from TDC to BDC and vice versa. The conformity between a strokes and processes is conditional, since most processes

For preliminary review

and strokes usually do not coincide in time. Typically, strokes are named by the name of the predominant process, for example: intake stroke, compression stroke, etc.

The *working stroke* – is the stroke of the piston at which useful work is done. In two-stroke engines, the working stroke is performed every revolution of the crankshaft, in four-stroke engines once every two revolutions.

Auxiliary strokes are piston strokes during which preparatory processes take place. They are carried out at the expense of kinetic energy of the moving parts of the crank mechanism or at the expense of working strokes of other cylinders (in a multi-cylinder diesel engine).

2.3 Ideal cycles of reciprocating engines

Thermodynamic analysis of the real cycle of a piston engine is quite difficult to perform, because during its course there are various losses that reduce the efficiency of heat utilisation compared to the theoretical cycle. Therefore, in engine theory, ideal closed thermodynamic cycles consisting of reversible thermodynamic processes are considered. Comparison of the efficiency values of the theoretical and actual cycles will make it possible to find out the degree of perfection of heat utilisation in a real engine.

The following assumptions are made when considering theoretical cycles:

– the same working medium is constantly used in the engine cycle, the quantity and composition of which does not change;

– the working medium is an ideal gas whose heat capacity does not depend on its temperature;

– the processes of compression and expansion of the working medium are considered adiabatic, i.e. it is assumed that the walls of the cylinder are heat-tight;

– the actual processes of fuel combustion and removal of combustion products (which occur in a real engine) in the theoretical cycle are conditionally replaced by heat input and output at constant volume (V = const) and pressure (p = const).

Ideal (thermodynamic or theoretical) cycles should be understood as a simplified thermodynamic model of the operating process. It is conventionally assumed that, except for the inevitable heat transfer to the cold source (according to the second law of thermodynamics), all other types of energy losses in such cycles are absent. Therefore, they are general theoretical engine cycles for any type of engine (four- or two-stroke), which simplifies the task of analysing actual internal combustion engine cycles.

As follows from the provisions and laws of technical thermodynamics, no premise of a theoretical cycle is realisable in practice. At the same time, a specific ideal cycle with the highest thermal efficiency is a benchmark for evaluating real working cycles of internal combustion engines.

There are three main types of theoretical cycles of reciprocating internal combustion engines (Fig. 2.7):

- with heat input Q_1, at constant volume V = const, which is the theoretical cycle of a petrol engine (Otto cycle) (Fig. 2.7 *a*);

- with heat input Q_2 at constant pressure p = const, is a theoretical cycle of compressor internal combustion engines, which are currently not built, but recently there is a return to this cycle in some medium-speed four-stroke diesel engines operating on heavy fuels (Diesel cycle) (Fig. 2.7 *b*);

- with heat input Q_1, and Q_2 at V =const and p = const, called the mixed heat input cycle. This cycle is the theoretical cycle of most modern internal combustion engines with compression ignition (Trinkler-Sabathé cycle) (Fig. 2.7 *c*).

Traditionally, piston engine cycles are represented in pV coordinates, which not only clearly reflect the character of pressure change as a function of volume change, but also give a visual representation of the work done per cycle. Taking into account that the work of the piston on the elementary section n will be equal to:

$$L_c^n = P_p S_p,$$

where P_p – is the force acting on the piston and S_p is the stroke of the piston at the n-th section of the cycle.

For Preliminary review

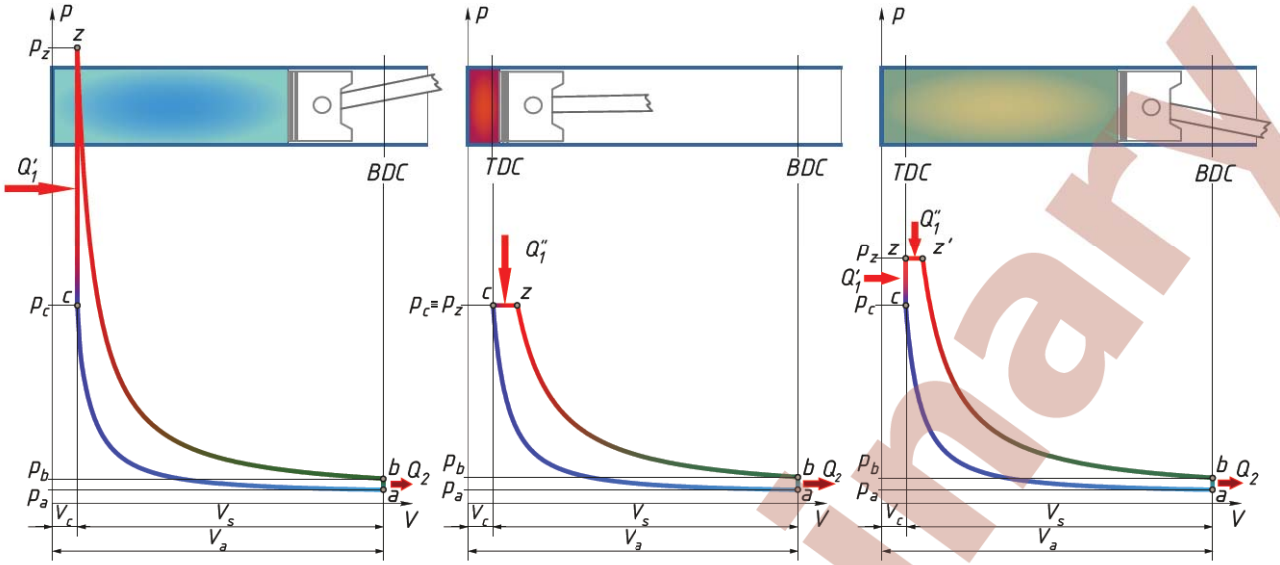


Figure 2.7 – Ideal piston engine cycles. *a* – Otto cycle; *b* – Diesel cycle; *c* – Trinkler-Sabathé cycle

The gas pressure force P_p is equal to the product of the average pressure in the cylinder p_p at the *n*-th section of the cycle by the piston area F_p

$$P_p = p_p F_p.$$

Substituting into the expression for the work at the *n*-th section of the cycle, we obtain the elementary work:

$$L_c^n = p_p F_p S_p.$$

Considering that $F_p S_p = V_s^n$, i.e. equal to the elementary volume at the *n*-th section of the cycle, finally the formula of work for the elementary section will take the form:

$$L_c^n = p_p V_s^n,$$

The total work of the cycle is equal to the sum of the work in the elementary sections:

$$L_c = \sum_1^n L_c^n$$

Thus, the area of the figure bounded by the processes constituting the cycle in pV coordinates will be equivalent to the work of the cycle.

The Trinkler-Sabathé* cycle impoverishes the features of the Otto* and Diesel cycles, so the main conclusions for this cycle are to some extent true for the other two cycles as well. Further we will consider these regularities in more detail.

In a cycle with mixed heat input, *as* – is the adiabatic compression of an ideal gas; *czz′* – is the heat input at V=const and p = const (in a real engine this corresponds to the process of fuel combustion), *z′b* – is the adiabatic expansion of the gas, *ba* – is the heat removal to a cold source at V= const.

The thermal efficiency of a mixed heat input cycle, all other things being equal, occupies an intermediate position between the first and second cycles. In general, it can be expressed as:

$$\eta_t = \frac{(Q_1' + Q_2'') - Q_2}{Q_1' + Q''} = 1 - \frac{Q_2}{Q_1' + Q''}$$

Expressing the heat input $Q_1 = Q'_1 + Q''_1$ and the heat output Q_2 through the temperatures at characteristic points of the cycle and the heat capacities and making the appropriate transformations, we obtain the formula of Seileger M.P.*:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\lambda\rho^k - 1}{\lambda - 1 + k\lambda(\rho - 1)} \qquad (2.1)$$

For preliminary review

where ε – is the degree of compression; $\lambda = p_z/p_c$ – is the degree of pressure rise; $\rho = V_z/V_c$ – is the degree of pre-expansion; k – is the adiabatic index of compression and expansion of gases.

The thermal efficiency of the Diesel cycle can be obtained from the above formula if we consider that in this cycle $\lambda = 1$

then:
$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \frac{\rho^k - 1}{[k\lambda(\rho - 1)]}.$$

The thermal efficiency of the Otto cycle can be obtained from the above formula by assuming $\rho = 1$

then:
$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}}.$$

The above dependences show that the thermal efficiency of the cycle invariably increases with increasing compression ratio.

Equation (2.1) allows us to trace the analytical relationship between the parameters of the mixed cycle.

It follows from the theory of technical thermodynamics that in order to increase the economy and efficiency of the thermodynamic cycle, it is necessary to expand the limits of changing the parameters of the working medium in the cycle. As the difference of gas parameters $\Delta T = T_{max} - T_{min}$, $\Delta p = p_{max} - p_{min}$; $\Delta V = V_{max} - V_{min}$ increases, the economy and efficiency of the cycle increase. This provision is fully applicable to engine duty cycles, although there are technical difficulties in their execution, since the values of ΔT and Δp are limited by the increase in thermal and mechanical stress, and ΔV by the increase in the size and mass of the plant.

As can be seen from equation (2.1), an increase in the degree of pressure rise λ leads to an increase in the thermal efficiency and an increase in the efficiency of the real engine operating process, while an increase in the degree of pre-expansion ρ – leads to a decrease in the cycle efficiency. Consequently, it is possible to influence the thermal efficiency η_t by varying the heat input Q'_1 and Q''_1. This allows to vary the temperature difference ΔT and the moment of onset of the expansion process. At ρ = 1.2...1.4 the thermal efficiency (η_t) is lower than at ρ = 1, but the maximum cycle pressure p_z, and consequently the mechanical stress of the engine is much lower. In a real engine, this results in lower mechanical losses and higher mechanical efficiency (η_m). Therefore, the effective engine efficiency (η_e), at ρ = 1.4 can be higher than at ρ = 1. Normal operation of diesel engine is provided at ρ = 1.3...1.4.

As previously mentioned, almost all modern internal combustion engines are supercharged. This makes it possible to increase the mass charge of air in the cylinders, which makes it possible to increase the cycle fuel supply and, consequently, to obtain greater effective power at the engine output flange.

The most widespread on marine diesel engines is gas turbine supercharging, where the compressor supplying air under increased pressure to the engine cylinders is driven by a gas turbine doing work at the expense of energy of gases exhausted in the engine cylinders. The experience of design and operation of combined engines has shown that the combination of a piston engine, compressor and gas turbine in one power unit, as well as intermediate cooling of charge air after the compressor before its supply to the cylinders has a positive effect on all thermal and gas-dynamic processes that make up the working cycle of a combined marine engine.

Gas turbine supercharging of modern internal combustion engines is performed according to two basic schemes: with pulse gas supply to the turbine (pulse supercharging) and with gas supply to the turbine at constant pressure (isobaric supercharging).

Both supercharging systems are shown schematically in Fig. 2.7. The same figure shows ideal cycles of combined engines with pulse and isobaric supercharging, represented in pV coordinates.

The ideal thermodynamic cycle is a simplified model of the operating process in which, apart from the inevitable heat transfer to the cold source, all other types of losses are neglected.

For preliminary review

The duty cycle of a combined engine is realised not only in the piston part, but also in the turbocharger unit. Let's consider in detail the ways of realisation and ideal cycles for each specific case. Since we have considered the operating process in the piston part of the engine earlier, let us dwell in detail only on the processes occurring outside the engine working cylinder.

In pulse boosting (Fig. 2.8 *a*), the exhaust gases in the cylinder, after opening the exhaust valve (ports), which corresponds to point *b* in the diagram, enter the gas turbine through a short, small-section connection pipe. Due to the small volume of the inlet pipe, there is practically no expansion of gases in it, so the pressure upstream of the turbine changes in the same way as in the working cylinder, i.e. in the form of a pulse.

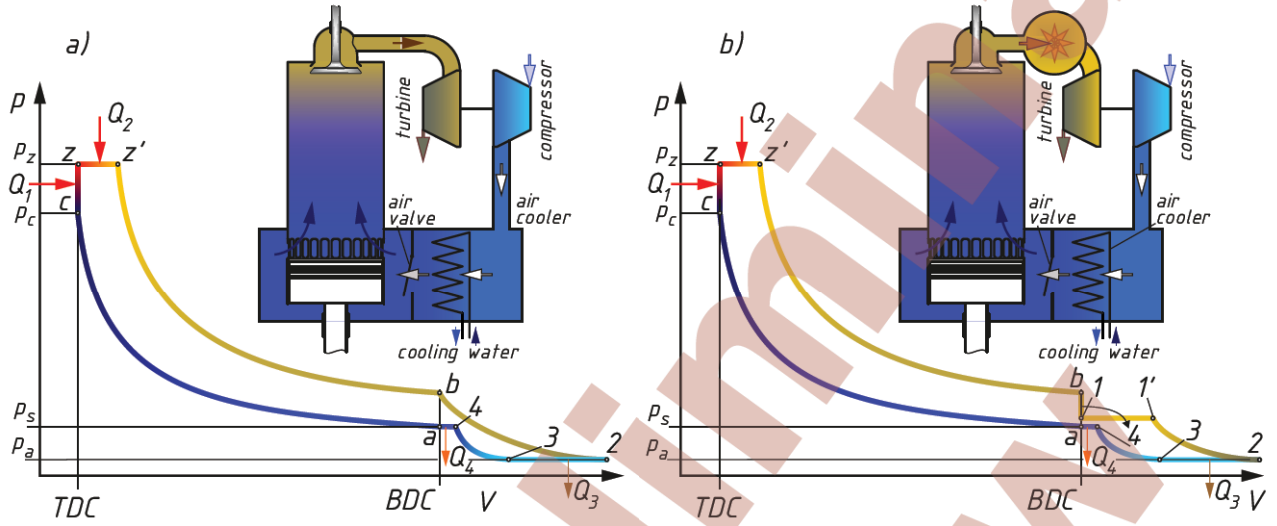


Figure 2.8 – Ideal cycles of a combined marine engine with pulse (*a*) and isobaric (*b*) supercharging

Further adiabatic expansion of gases occurs already in the flowing part of the gas turbine, as a result of which the impulse energy of gases is converted into mechanical work. In the indicator diagram, this process corresponds to section *b-2*. Further exhaust gases at constant pressure (p = const) are discharged into the atmosphere or utilisation system taking with them a part of heat Q_2 (heat transfer to a cold source according to the second law of thermodynamics). On the indicator diagram this process corresponds to section 2-3. Mechanical energy received in the gas turbine is fed to the air compressor, where preliminary adiabatic compression of the working medium (atmospheric air) takes place (process 3-4). As a result of adiabatic compression, the temperature of the working medium at the compressor outlet rises to 150...200°C.

In order to increase the density of the working medium and reduce thermal loads on engine parts, the air is pre-cooled in special heat exchangers called air coolers to a temperature of 40...50°C (heat Q_3) before entering the working cylinder. This process takes place at constant pressure (p = const), and on the indicator diagram it corresponds to section *4-a*. Further, the pre-compressed and cooled working medium through the air valve enters the working cylinder (point *a*) where processes similar to those in a non-injected piston engine take place.

At isobaric supercharging (Fig. 2.8 *b*) the exhaust gases in the cylinder after opening the exhaust valve (ports), which corresponds to point *b* on the indicator diagram, enter the gas receiver, the volume of which is 15...20 times higher than the volume of the working cylinder. When gases enter the receiver, their pressure decreases sharply at V = const (process *b-1*). Part of the kinetic energy of the exhaust gases in the exhaust receiver is converted into thermal energy, which increases the temperature of the gases in front of the gas turbine (the energy transfer is conventionally shown by the arrow). The large volume of the receiver smoothes out individual pressure pulses, resulting in a constant pressure in the receiver (p=const). On the indicator diagram this process corresponds to the section 1-1′. Further gases at constant pressure go to the gas turbine, where they expand according to adiabatic (section 1′-2). Further processes of working medium preparation proceed in the same way as at pulse supercharging.

Preliminary review

Isobaric supercharged engines are becoming more and more widely used in the fleet, so below we will consider in more detail the influence of the main factors on the efficiency of the thermodynamic cycle of such an engine.

In addition to the relationships (ε, λ, ρ) already discussed for the non-injected engine, the following relationships should be added:

$\varepsilon_1 = V_3/V_4$ – is the compression ratio in the compressor;

$\varepsilon_{ce} = \varepsilon_1\varepsilon$ – is the total compression ratio of the combined engine;

$\omega = V_4/V_a$ – degree of volume reduction with intercooling. In the absence of intercooling $V_4 = V_a$ and ω=1;

$\varphi_1 = V_a/V_1$ – degree of gas volume change (practically in four-stroke diesels φ_1 =1, in two-stroke diesels φ_1 =0,9...0,8);

$\tau = p_a/p_1 = p_s/p_t$ – is the degree of pressure reduction (in the diesel exhaust pipe before the turbine);

$\pi_k = p_4/p_3 = p_s/p_0$ – degree of pressure rise (in the compressor);

z – is the degree of pressure reduction (in the turbine); since $p_1 = p_{1'}$; $p_4 = p_a$; $p_3 = p_2$, then $z = \pi_k/\tau$.

Taking into account the given relations and equalities, let us write down the thermal efficiency of the combined engine cycle in a general form:

$$\eta_t = 1 - \frac{Q_3 + Q_2}{Q_1' + Q_1''}$$

Substituting into this equation the heat values represented through the cycle parameters, we obtain in expanded form an expression for the thermal efficiency, which allows us to draw conclusions about the influence of individual parameters on the cycle efficiency of the combined engine:

$$\eta_t = 1 - \frac{k\varepsilon_1^{k-1}(\omega - 1) + \frac{\tau^{(k-1)/k}}{\varphi_1^{k-1}}\left[\lambda\rho^k + \frac{k-1}{\tau}\varphi_1^k\right] - k\omega}{\varepsilon_{ce}^{k-1}\left[\lambda - 1 + k\lambda(\rho - 1)\right]}. \qquad (2.2)$$

Having analysed equation (2.2) the following conclusions can be drawn:

- the thermal efficiency of the cycle with intermediate air cooling is lower than in the cycle without cooling. In this case, the difference between the efficiency values $\Delta\eta_t = \eta_t - \eta_{tcol}$ increases with increasing degree of pressure rise in the compressor π_k. In a real engine, lowering the charge air temperature by intercooling it reduces the effective efficiency less than the thermal efficiency in the theoretical cycle. Intercooling allows to increase the specific work of the cycle without increasing its maximum temperature;

- the thermal efficiency of the cycle with constant gas pressure before the turbine is about 4...8% lower than the thermal efficiency of the cycle with pulse supercharging. As the degree of pressure increase in the compressor increases (π_k), the difference between the thermal efficiency values of the cycles decreases. Thus, the use of pulse systems is only effective at low boost pressures. As the degree of pressure increase in the compressor increases, the use of pulse systems becomes irrelevant.

- increase of the working medium expansion limits using gas turbines and increase of the total compression ratio of the combined engine (ε_{ce}) due to pre-compression in the compressor, allows to increase the thermal efficiency of the cycle. So at compression ratio in the working cylinder ε = 12...13 and the degree of pressure increase in the compressor π_k = 2...3 for the combined engine η_t = 0.7...0.72;

in contrast to the thermodynamic cycle, additional energy losses occur in the operating process, reducing its efficiency, which is evaluated by the indicator efficiency η_i. Indicator efficiency characterises the degree of perfection of transformation of thermal energy into mechanical work inside the engine cylinder. In modern marine diesel engines it is more than 50 %.

For preliminary review

2.4 Real cycles of marine diesel engines

The four-stroke engines used in the navy have predominantly valve timing with the valves mounted on top of the cylinder head. The valves are held closed by springs and by the pressure of the gases in the cylinder. The valves are opened at the right moments by a timing mechanism driven by the engine crankshaft. Each valve is opened once per two revolutions of the crankshaft.

For both ideal and real cycles, the engine operating processe is traditionally represented in *pV*-coordinates. For a real cycle, auxiliary processes associated with the need for gas exchange between the working cylinder and the environment are additionally plotted on the diagram. Schematic representation of the four-stroke diesel engine operation order combined with indicator diagrams and diagrams of gas distribution phases is presented in Fig. 2.8.

Timing diagrams – represent the duration of the processes occurring in a real engine, which are represented in the polar coordinate system as a function of the crankshaft rotation angle. The pole of the diagram is conventionally aligned with the crank rotation axis, and the processes are represented as circular arcs.

2.4.1 Operating procedures for a four-stroke diesel engine

The first stroke* is *intake (Fig. 2.9, *a*). When the crankshaft rotates (in the direction of the arrow), the piston moves from TDC to BDC, and the distributor mechanism opens the intake valve 1, and communicates the cylinder supra-piston space with the intake pipe.

In a real engine, the intake process consists of three stages.

The first stage (I) is the dynamic purging stage. The inlet valve opening starts 50...80° before the piston reaches TDC (point *r* on the indicator diagram), this interval is called the *inlet valve opening advance angle* (φ_{Iiv}). This is done to ensure that the cylinder is filled to its maximum capacity with air, so that there is sufficient air flow through the cylinder by the time the piston starts to move to TDC. The combustion chamber volume is dynamically purged by the energy of the exhaust gases and the air pressure in the intake receiver.

The second stage (II) is the mechanical suction stage. The piston moves from TDC to BDC, increasing the cylinder displacement. Through the open inlet valve, the turbocharger (TC) supplies air into the super-piston space under pressure p_k =0.13...0.45 MPa.

The third stage (III) is the stage of dynamic supercharging. After the piston arrives at BDC, the intake valve does not close immediately, but remains open for another 30...50°. This angular interval is called the *intake valve closing lag angle* (φ_{IIIiv}). During this interval of the cycle, the cylinder is dynamically charged with air due to the inertia of the gas flow in the intake pipe, the velocity of which can reach 70 m/s by the end of intake.

On the indicator diagram the intake process corresponds to the line *ra*. At the end of the filling process, the air in the cylinder has the following parameters:

$$p_a = 0.130...0.45 \text{ MPa}; \; t_a = 40...130°\text{C}.$$

The second stroke is compression. The piston moves from BDC to TDC, compressing the air charge (Fig. 2.9, *b*).

In a real engine, compression starts from the moment of intake valve closure (point *a* on the indicator diagram) and continues until the piston reaches TDC. By the end of the compression process, the charge temperature must ensure stable autoignition of the fuel, for this purpose it must exceed the autoignition temperature of the fuel by 160...200°C.

An ideal compression process (line 1 in Fig. 2.10) should be adiabatic and described by the equation pV^{k_1} = const, where k_1 – is the compression adiabatic exponent. In the real compression process, there are always leakages of charge through leakages in the working space and heat exchange with the cylinder walls.

Indicator efficiency takes into account not only heat removal to the cold source, but also heat losses in the working cylinder. It characterises the degree of perfection of conversion of heat energy into mechanical work inside the engine cylinder.

For preliminary review

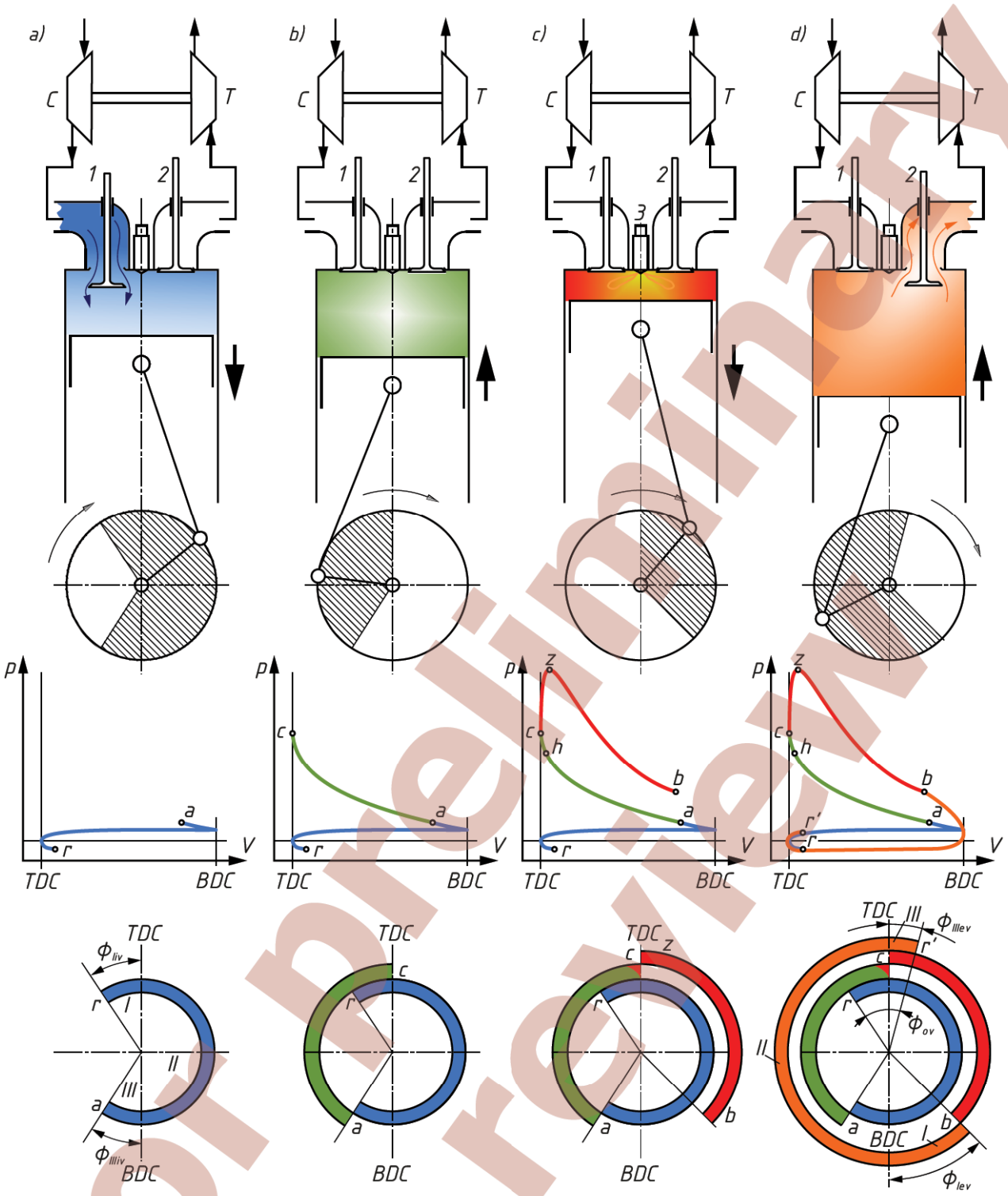


Figure 2.9 – Operating process of a four-stroke diesel engine, indicator diagrams of the real operating cycle and diagrams of gas distribution phases: *a* – first stroke – intake; *b* – second stroke – compression; *c* – third stroke – combustion and expansion (working stroke); *d* – fourth stroke – discharge

As a result of leakages, there is, though insignificant, but still a loss of charge, and the heat exchange of charge with the walls is of alternating character. Therefore, the actual compression process is polytropic.

At the initial stage of compression, the charge temperature is lower than the wall temperature and heat is supplied to the charge from them. In this case, the polytropy of compression n_1 is higher than the adiabatic index, and the polytropy of compression (line 1 in Fig. 2.10) is steeper than the adiabatic (line 2).

For preliminary review

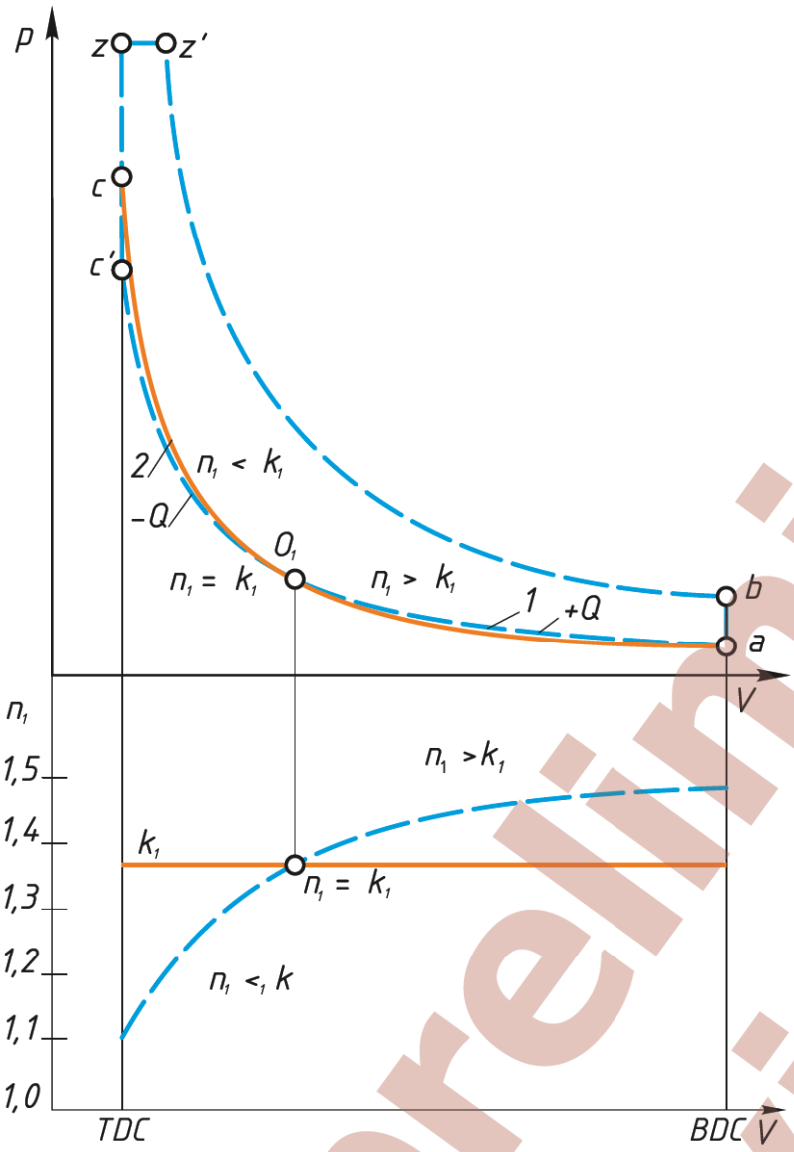


Figure 2.10 – Diagrams of actual and calculated processes of charge compression in the cylinder

As the charge is compressed, its temperature increases and the heat transfer from the walls to the charge decreases. At some point, the charge and wall temperatures become equal, there is no heat transfer at this point, and $n_1 = k_1$. At further compression the charge temperature becomes higher than the average wall temperature and heat starts to be transferred from the charge to the walls. The compression polytropy index n_1 appears to be lower than the adiabatic index and its curve follows a more gentle curve with respect to the adiabatic. As a result, the pressure at the end of compression in the actual cycle (point c) will be lower than the pressure at the end of adiabatic compression (point c'). In marine diesel engines, the compression polytropy n_1 of the actual cycle varies in the range of 1.5...1.1. The first value corresponds to the beginning of the process, the second – to the end.

In operational practice, to estimate the charge parameters during the compression process, a simplified method is used, where the assumption is made that the compression process is polytropic with a conditional constant mean n_1. In this case, the total amount of heat communicated to the charge during the compression process is equal to zero ($\Delta Q = 0$), i.e. the compression process is considered as pseudoadiabatic. The average value of the polytropy index n_1, is accepted for medium speed engine (MSE) and high speed engine (HSE) within the limits $n_1 = 1.38...1.4$, for low speed engine (LSE) – $n_1 = 1.34...1.42$.

Pressure and temperature of the charge at the end of compression can be determined as a result of transformation of the compression polytropy equation. For initial and end points of compression the equation of polytropy with constant exponent n_1 has the form:

For preliminary review

$$p_a V_a^{n_1} = p_c V_c^{n_1} \quad (2.3)$$

where p_a, p_c – pressure at the beginning and end of compression, kPa; V_a, V_c – cylinder volume at the beginning and end of compression, m^3.

As a result of transformation of the equation, we obtain a formula for determining the pressure of the charge at the end of compression:

$$p_c = p_a \times (V_a / V_c)^{n_1} = p_a \times \varepsilon_0^{n_1}$$

where ε_0 – is the actual compression ratio.

The charge temperature at the end of compression is determined as a result of the joint solution of the polytropy equation and the equations of state of the charge at points *a* and *c* (see Fig. 2.10) of the compression process:

$$p_a V_a = R_\mu G_a T_a ;$$
$$p_c V_c = R_\mu G_a T_c ;$$

where R_μ – universal gas constant (R_μ = 8.31 kJ/(kmol×K)); G_a – mass of the cylinder charge, kmol; T_a, T_c – temperature of the charge at the beginning and end of compression, K.

After transformations of equation (2.3) taking into account the above equations, we obtain a formula for determining the charge temperature at the end of compression:

$$T_c = T_a \times (V_a / V_c)^{n_1 - 1} = T_a \times \varepsilon_0^{n_1 - 1} .$$

In modern four-stroke internal combustion engines the pressure and temperature at the end of compression are within the limits p_c =4.5...16.0 MPa, t_c =530...930°C. The compression process on the indicator diagram (Fig. 2.9, 2.10) is depicted by the line *as*.

The processes of mixing, ignition and combustion take some time. For efficient utilisation of heat released during combustion, it is necessary that fuel combustion ends at piston position close to TDC. Therefore, fuel injection into the combustion chamber is performed with the help of injector 3 (Fig. 2.9) under pressure of 100...250 MPa 5...30° before the piston comes to TDC (point *h* on the indicator diagram). This cycle interval is called the *fuel injection advance angle* (φ_{pdt}).

The third stroke is combustion and expansion (working stroke). The third stroke takes place during the piston stroke from TDC to BDC (Fig. 2.9, c). During this stroke, useful work is performed in the diesel engine – conversion of gas thermal energy into mechanical work of the piston.

At the beginning of the stroke, the fuel that entered the cylinder and was prepared for combustion at the end of the second stroke burns intensively. As a result of the release of a large amount of heat, the temperature and pressure in the working cylinder rise sharply; despite some increase in volume (section *cz* on the indicator diagram). By the end of combustion, the pressure in the cylinder reaches p_z –8...23 MPa and the temperature of gases t_z –1450...1730°C. Under the action of pressure there is a further movement of the piston to the BDC and expansion of gases. During expansion gases do useful work. The process of gas expansion on the diagram (Fig. 2.9) corresponds to the section *zb*.

Charge expansion starts from the moment the piston starts to move away from TDC. At the first stage it occurs when fuel combustion is still going on in the cylinder, which is accompanied by molecular changes in the composition of the working medium, high pressures and temperatures. The latter causes intensive heat exchange between the working medium and the walls of the working space. After the end of visible combustion begins the expansion of combustion products, which is also accompanied by a number of physical and chemical processes, the main of which is the transition of internal energy of gases in the mechanical work of the piston. At the same time, the pressure and temperature of the charge decrease, heat exchange between the working medium and the walls of the working space, afterburning of fuel, recovery of dissociation products, loss of charge through the gas joint leakage, etc. occur.

The combined action of the above factors at the expansion site determines the polytropic charac-

For preliminary review

ter of the process with the index n_2 (purple dashed line – Fig. 2.11), which varies depending on the intensity of the prevailing factors throughout the process.

In the first part of the process (up to the point p_z), the dominant factors are heat input, molecular changes and intensive heat exchange with the cylinder walls. In this section, the value of the expansion polytropy index n_2 is lower than the expansion adiabatic index k_2, in some cases even less than 1. It should be noted that molecular change in the composition of the working medium leads to a decrease in the adiabatic index, until the process of fuel afterburning is completely completed on the expansion line at point 2 (Fig. 2.11). As the piston moves to the BDC, the process of fuel afterburning becomes less intensive, and the heat exchange with the walls increases due to the increase in the cooling surface. At this point, the polytropic index of expansion increases continuously.

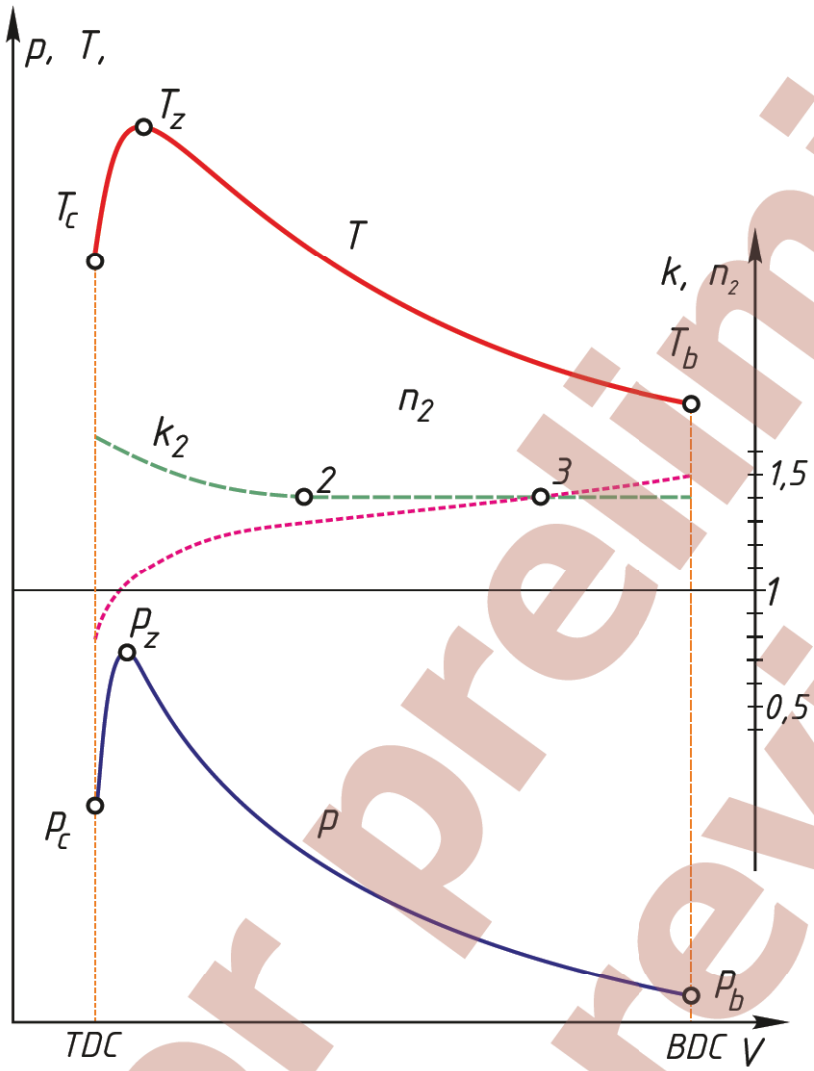


Figure 2.11 – Change in the process of expansion of indicators T, p, n_2 and k_2

At a certain position of the piston, the heat input due to afterburning of fuel and recovery of dissociation products becomes equal to the heat transfer to the cylinder walls. The current value of the polytropy index reaches the value of the adiabatic index, $n_2 = k_2$ (point 3 – instantaneous thermal equilibrium of the state of gases and cylinder walls). At further expansion the heat dissipation into the cylinder walls becomes the dominant factor, and the index n_2 becomes higher than the adiabatic index k_2. As the piston approaches the BDC n_2 increases to 1.4...1.5. Thus, in a real engine, the expansion process after completion of visible combustion is carried out by polytrope with a variable index from 1.1 to 1.5.

Pressure and temperature of the end of expansion. *The end expansion* parameters (pressure p_b and temperature T_b of the gases) depend on the maximum combustion parameters $p_{z'}$, $T_{z'}$, the degree of subsequent expansionδ and the expansion polytropy n_2:

For preliminary review

$$p_b V_b^{n_2} = p_{z'} V_{z'}^{n_2},$$

from where

$$p_b = p_{z'} / (V_b / V_{z'})^{n_2} = p_{z'} / \delta^{n_2}. \quad (2.4)$$

Then:

$$T_b = T_{z'} / \delta^{n_2 - 1}, \quad (2.5)$$

where δ is the degree of subsequent expansion $\delta = V_b / V_{z'}$.

When estimating the operating process parameters, it is assumed that the expansion process starts at the end of visible combustion at the point *z'*, resulting in a polytropic decrease in the pressure and temperature of the gases. Due to the difficulty of taking into account all acting factors accompanying the expansion process, an averaged constant polytropy index n_2 is used to determine the parameters of gases. The choice of a constant index is based on the condition that in the process of expansion with a constant index n_2 the work obtained will be the same as in the actual expansion with a variable polytropy index.

In a valid cycle, expansion ends at the moment the exhaust ports or valves open.

Based on the experimental data, the average expansion polytropy p_2 for low- and medium-speed marine engines is within the range of 1.20...1.30, for high-speed engines – 1.10...1.25. Pressures p_b and temperatures T_b at the end of expansion for marine internal combustion engines are given in Table 2.1.

Table 2.1 – Pressures and temperatures at the end of expansion of marine internal combustion engines

Engine type	p_b, MPa	T_b, K
Low speed engines	0.25...0.85	900...1000
Medium speed engines	0.30...0.90	900...1100
High speed engines	0.40...0.95	1100...1200

The fourth stroke is exhaust. During the fourth stroke, the cylinder is cleared of exhaust gases (Fig. 2.9, d). The piston, moving from BDC to TDC, displaces gases from the cylinder through the open exhaust valve 2.

In a real engine, the intake process consists of three stages.

The first stage (I) is the free exhaust stage. The exhaust valve starts to open at the moment when the piston does not reach the BDC 40...60°. This section of the cycle is called the exhaust *valve opening advance angle* (φ_{Iev}). At this time, the pressure of gases in the cylinder is still high enough and gases under excessive pressure go to the exhaust manifold. This reduces the resistance to the piston movement from BDC to TDC, improves cylinder cleaning and increases the power of the turbocharger's gas turbine.

The second stage (II) is the mechanical displacement stage. The piston moves from BDC to TDC, pushing gases out of the cylinder, through the open exhaust valve.

The third stage (III), the dynamic purge stage, overlaps with stage *I* of the intake process. The stage starts at TDC and ends after the exhaust valve closes 40...70° after TDC. This section of the cycle is called the exhaust valve *closing lag angle* (φ_{IIIev}). The timing diagram shows that the end of the exhaust stroke and the beginning of the intake stroke overlap. The part of the cycle where both valves are open is called the valve overlap *angle* (φ_{ov} = 25...140°). During this cycle interval, the gases leaving the exhaust manifold at a high velocity create additional fresh air flow through the piston space via the open intake valve. As a result, the cleaning of the combustion chamber is improved and the temperature of its walls is reduced.

On the indicator diagram, the *br'* line corresponds to the fourth beat.

The fourth cycle ends the working cycle. With further piston movement, all cycle cycles are repeated in the same sequence.

For preliminary review

2.4.2 Operating procedures and real cycle of a marine two-stroke diesel engine

In two-stroke diesel engines, the operating cycle is completed in one revolution of the crankshaft. In this case, the process of gas exchange occurs when the piston moves near the BDC. Cleaning of the cylinder from exhaust gases is carried out by displacing them with air, pre-compressed to a pressure of 0.15...0.33 MPa, which enters the cylinder through purge ports placed in the lower part of the cylinder. Pre-compression of air is carried out in a special purge pump or compressor, which is a separate unit. The exhaust gases are discharged through exhaust ports cut into the cylinder liner (slotted blowdown) or through an exhaust valve (valve-slotted blowdown).

Fig. 2.12 shows the scheme of operation of a two-stroke marine engine with a direct-flow, valve-slotted gas exchange scheme, which is predominantly used in modern internal combustion engines.

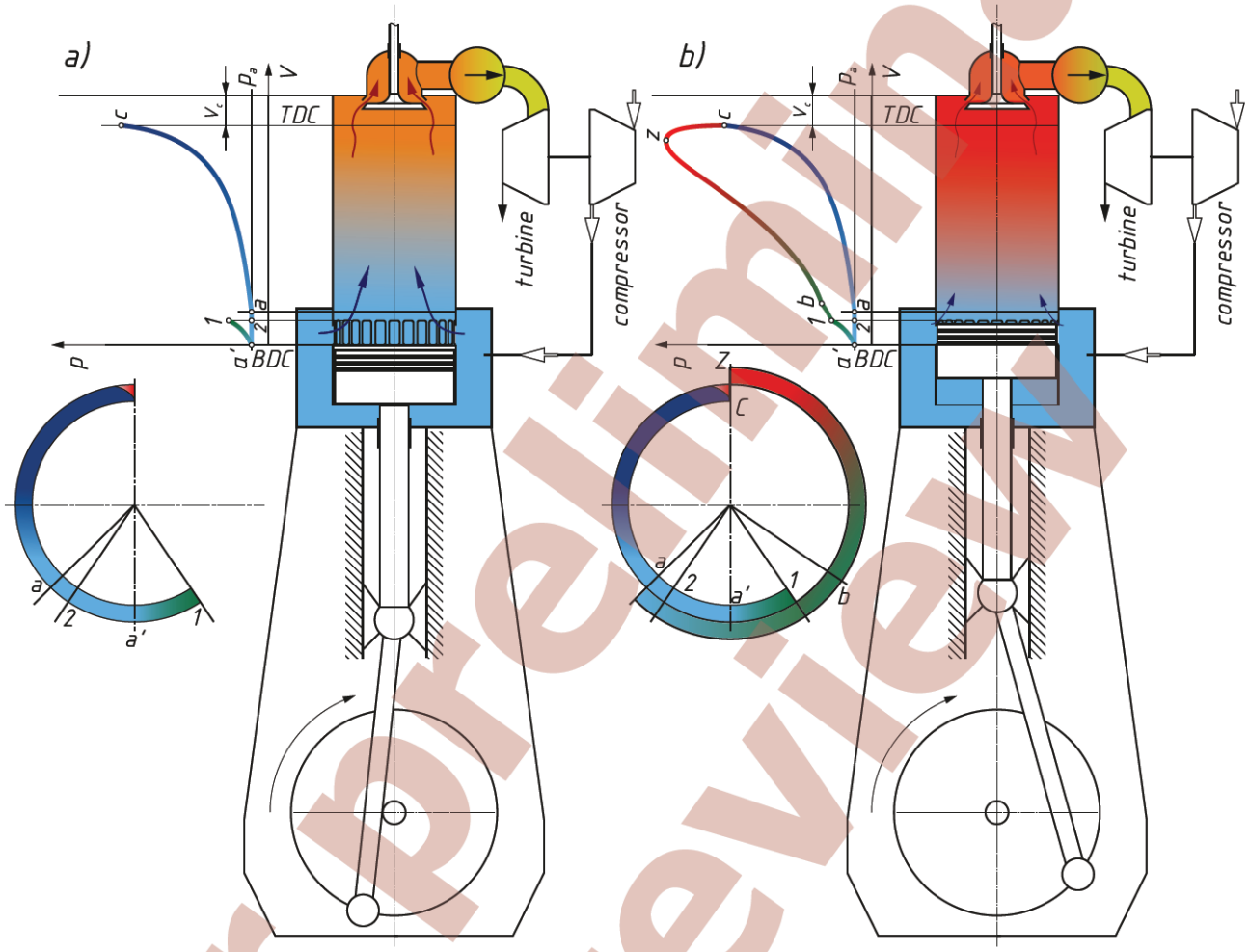


Figure 2.12 – Operating process of a two-stroke diesel engine, indicator diagrams of the real operating cycle and diagrams of gas distribution phases: *a* – first stroke – blowdown and compression; *b* – second stroke – combustion, expansion, exhaust and blowdown

The main features of engines made according to this scheme are:

- intake ports, located in the lower part of the cylinder along the entire perimeter. The height of the ports is 5...10% of the piston stroke;
- the exhaust valve, is located in the cylinder head and its actuator ensures that the valve is opened at each revolution of the crankshaft;
- cylinder purging at medium and full power modes is carried out with air pre-compressed in the turbocharger unit, and at start-up and low load modes with air compressed in externally driven blowers.

The operating cycle in the engine is carried out as follows.

The first stroke is purging and compression (Fig. 2.12, *a*). As the piston moves towards BDC, the purge ports open and air enters the cylinder from the charge air receiver (point 1 on the indicator

For preliminary review

diagram). The stroke begins as the piston moves from the BDC (point *a'*) to the TDC. At the beginning of this stroke, the piston continues to purge and release combustion products through the open exhaust valve. When the piston closes the purge ports, the air flow into the working cylinder stops (point 2) and until the exhaust valve closes (point *a*), some part of the charge is displaced into the exhaust receiver. After closing the gas exchange organs, the compression process begins (line *as*), which ends when the piston arrives at TDC. By the end of compression (point *c*), the pressure in the cylinder increases to p_c =8.5...16.5 MPa, and the temperature T_c = 530...830 °C.

The second stroke – combustion, expansion, exhaust and purging (Fig. 2.12, *b*) – is performed during the piston stroke from the TDC to the BDC. Depending on the type of diesel engine, 3...35°. before the piston arrives at TDC, fuel is injected into the cylinder, which ignites and burns itself (combustion process – line *cz*). As a result, the pressure in the cylinder increases to p_z =11.5...18.5 MPa, and the temperature T_z = 1200...1700 °C. Under the action of the gas pressure the piston in the process of movement from TDC to BDC makes a working stroke, (line *zb*) In the process of expansion the temperature and pressure of gases decrease. 80...90° before BDC, the exhaust valve opens (point *b*) and the combustion products start to be discharged from the cylinder, while the pressure in the cylinder drops sharply. The exhaust gases flow into the gas turbine.

The purge ports are opened by the piston after opening the exhaust valve, when the pressure in the cylinder becomes slightly lower than the air pressure entering the cylinder from the purge receiver. When the purge ports open, the cylinder begins to purge and fill with fresh charge. Purging continues until the piston closes the purge ports as it moves towards TDC.

Thus, in two-stroke engines, there is no clear boundary between the individual cycles; the processes of fresh charge intake and exhaust gas discharge overlap and occur almost simultaneously.

The above-mentioned straight-through valve-slotted gas exchange scheme is now predominantly used on modern powerful marine low-speed diesel engines. All leading world manufacturers of internal combustion engines have switched to this scheme. However, there is still a large number of engines in operation with various slotted schemes of working cylinder purging, therefore, the features of gas exchange processes in engines of this type will be considered further (Fig. 2.13).

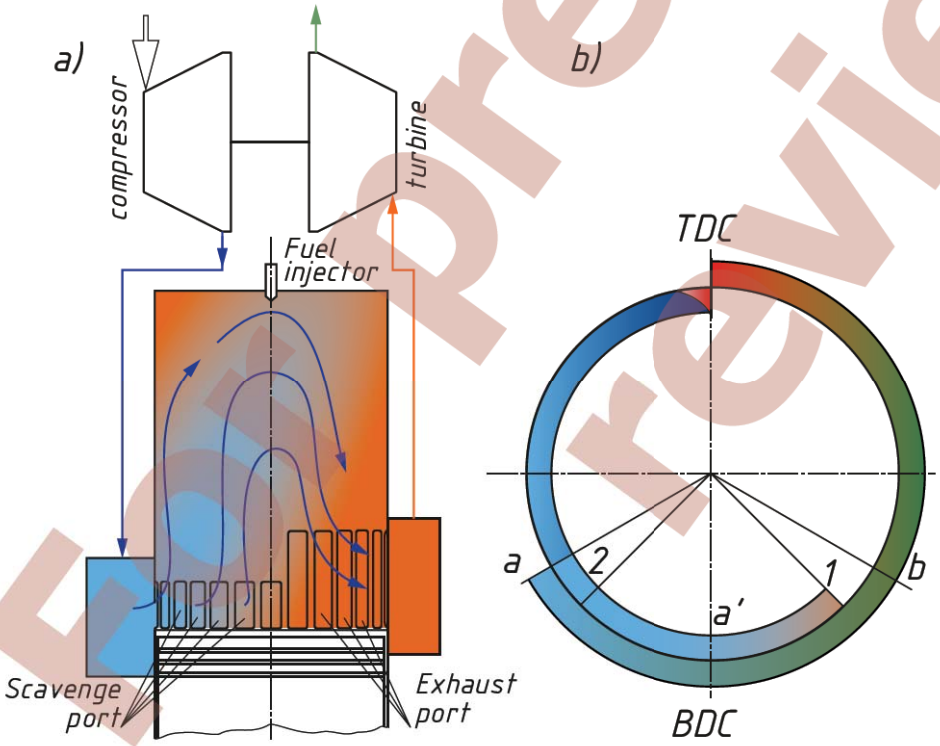


Figure 2.13 – Gas exchange process of a two-stroke diesel engine with slotted blowdown scheme (*a*) and gas distribution phase diagram (*b*)

The main features of such engines are:

- two types of ports, exhaust and purge, which can be located on top of each other or on opposite cylinder walls;
- the upper edge of the exhaust ports lies at a height of 30...40% of the piston stroke;

For preliminary review

- the exhaust and intake process is controlled by the piston itself as it moves around the BDC.

In this type of engine, the piston, as it moves from TDC to BDC, first opens the exhaust ports, through which gases enter the exhaust receiver and then the turbine. As the piston moves further, the purge ports open and air pre-compressed in the turbocharger enters the cylinder. The air supply is organised in such a way that the exhaust gases are cleared from the cylinder as much as possible.

The timing of these engines is symmetrical with respect to the TDC and cannot be changed during engine operation.

The advantages of this purging scheme are:

- simplicity of engine design, which is expressed in the absence of a complex system of valves and their drive, as well as a simpler design of the working cylinder head;
- the possibility of centralised injectors positioning, which promotes quality mixing.

The disadvantages of engines of this design include:

- significant loss of operating stroke due to the high height of the exhaust ports;
- impossibility of quality cleaning of the working space from exhaust gases. There are always areas in the cylinder that are practically free of fresh charge;
- the need to have sufficient piston height to avoid simultaneous opening of the purge and exhaust ports when the piston is close to TDC;
- high charge losses due to the fact that from the moment of closing the purge ports to the moment of closing the exhaust ports, the charge is displaced by the piston into the exhaust receiver.

For the above reasons, which do not allow further forcing of the operating process, the slotted purge schemes are now abandoned. The last manufacturer of slotted purge engines was the Swiss company Sulzer, for which this engine layout was traditional. However, since 1981, the company has started production of new RTA series engines, which use a straight-through valve-slotted purge arrangement.

2.5 Main engine performance indicators

2.5.1 Indicator indicators

As it was shown earlier, the work done by gases in the working cylinder is equivalent to the area of the indicator diagram. In practice, it is rather difficult to consider the area of the curvilinear diagram, so for simplicity of processing it is represented in the form of a rectangle, one side of which is equal to the working volume V_s, and the other side corresponds to some average pressure, under the action of which the piston during one working stroke does work equivalent to the indicator work of a real closed cycle. Such conditional pressure has received the name of *average indicator pressure* – p_i, and the work done per cycle, *indicator work* – L_i.

To find p_i, the area of the indicator diagram is represented as the sum of rectangles equal in area to the elementary fragments of the diagram, the curvature of which is neglected. The total indicator work of the cycle L_i is found as the sum of elementary works, kJ (Fig. 2.14):

$$L_i = \sum_{n=1}^{n=20} L_n ,$$

and the average indicator pressure will be equal to, kPa (Fig. 2.15),

$$p_i = L_i / V_s. \qquad (2.6)$$

If we refer the work of the cycle to the working volume of the cylinder in m^3, we obtain the specific work of the actual cycle in the form:

$$p_i = \frac{L_i}{10^3 V_s}, \text{ kJ/m}^3.$$

Thus, p_i defines the cycle work attributed to one m^3 of cylinder displacement and is independent of its geometrical dimensions.

For Preliminary review

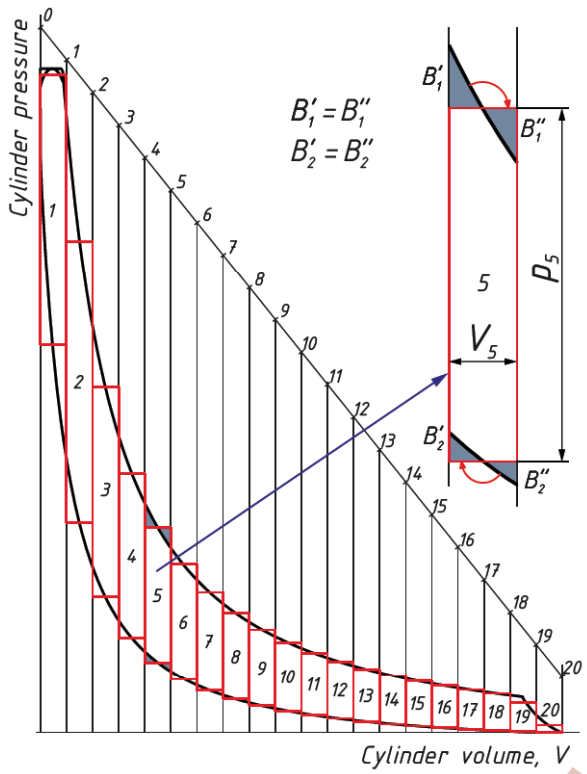


Figure 2.14 – Determination of indicator cycle work of piston engine

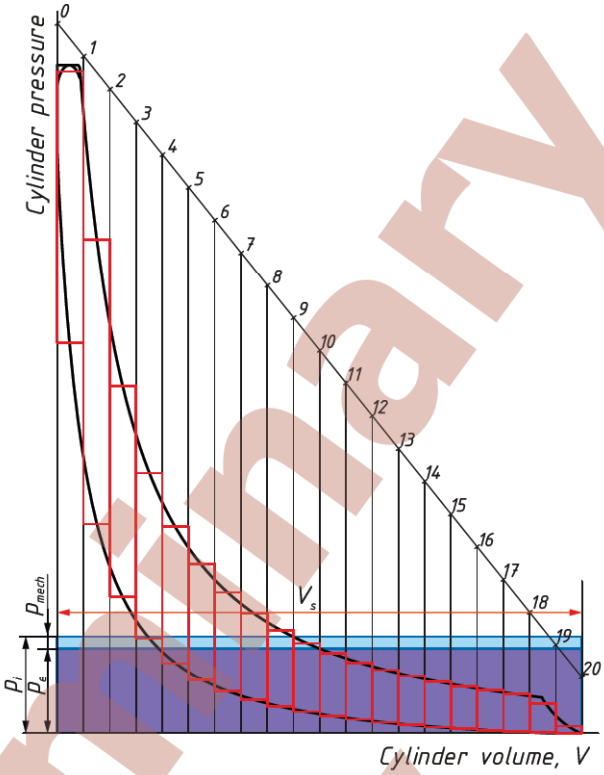


Figure 2.15 – Determination of average indicator pressure, average effective pressure and mechanical loss pressure

Let us find the useful indicator work of gases L_i' for the theoretical cycle with mixed heat supply (Fig. 2.16) as an algebraic sum of indicator works of separate processes:

$$L_i' = L_{zz'} + L_{z'b} - L_{ac}\,, \tag{2.6}$$

where $L_{zz'}$ – is the work of gases in the process of heat input at p = const; $L_{z'b}$ – is the work of gases in the process of expansion at n_2 = const; L_{ac} – is the work of gases in the process of compression at n_1 =const.

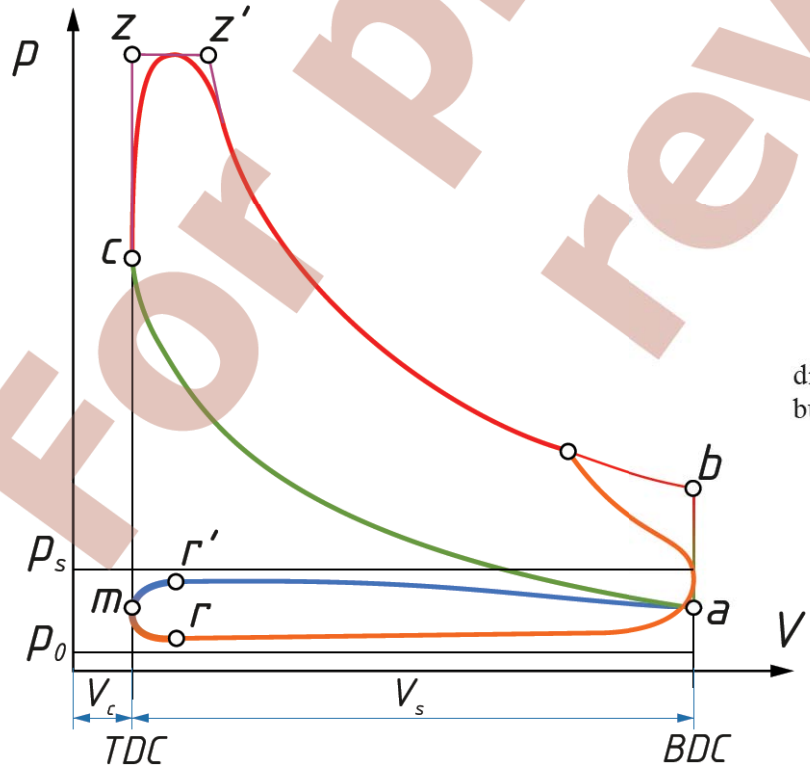


Figure 2.16 – Theoretical indicator diagram of a four-stroke internal combustion engine

For preliminary review

In the processes *cz* of heat input and *ba* of heat output, which occur without volume change (V=const), no work is done.

For the processes $L_{zz'}$, $L_{z'b}$ and L_{ac} we express the work through the cycle parameters (pressures and volumes):

$$L_{zz'} = p_z(V_{z'} - V_c);$$

$$L_{z'b} = \frac{p_z V_{z'} - p_b V_b}{n_2 - 1};$$

$$L_{ac} = \frac{p_c V_c - p_a V_a}{n_1 - 1};$$

substituting into (2.6), we obtain:

$$L_i' = p_z(V_{z'} - V_c) + \frac{p_z V_{z'} - p_b V_b}{n_2 - 1} - \frac{p_c V_c - p_a V_a}{n_1 - 1}.$$

Substituting the obtained expression into formula (2.6), replacing V_s by $V_c(\varepsilon - 1)$ and taking $1/(\varepsilon - 1)$ out of brackets, we obtain an expression for determining the *mean design indicator pressure*:

$$p_i' = \frac{1}{\varepsilon - 1}\left[p_z\left(\frac{V_{z'}}{V_c} - 1\right) + \frac{1}{n_2 - 1}\left(p_z \frac{V_{z'}}{V_c} - p_b \frac{V_b}{V_c}\right) - \frac{1}{n_1 - 1}\left(p_c - p_a \frac{V_a}{V_c}\right)\right].$$

Let us replace the parameters and expressions in square brackets by known relations:

$$p_z = \lambda p_c; \quad \frac{V_{z'}}{V_c} = \rho; \quad p_b = \frac{\lambda p_c}{\delta^{n_2}}; \quad \frac{V_b}{V_c} = \varepsilon; \quad p_a = \frac{p_c}{\varepsilon^{n_1}}; \quad \lambda\varepsilon = \lambda\rho\delta$$

and take p_c out of the brackets. As a result, we obtain the final expression for the mean design indicator pressure of the theoretical cycle with mixed heat supply

$$p_i' = \frac{p_c}{\varepsilon - 1}\left[\lambda(\rho - 1) + \frac{\lambda\rho}{n_2 - 1}\left(1 - \frac{1}{\delta^{n_2 - 1}}\right) - \frac{1}{n_1 - 1}\left(1 - \frac{1}{\varepsilon^{n_1 - 1}}\right)\right]. \qquad (2.7)$$

It follows from expression (2.7) that p'_i is directly proportional to p_c and is functionally dependent on the dimensionless duty cycle parameters ε, λ, ρ, δ, and n_1 and n_2 and is independent of the main cylinder dimensions.

The actual average indicator pressure p_i in contrast to p'_i is less than the calculated one, due to the difference of the theoretical cycle diagram from the working cycle diagram at the combustion process section *czz'* and at the end of expansion (point *b*), as well as due to the deviation of the actual compression and expansion processes from the polytropic ones.

These deviations are accounted for by the indicator diagram rounding factor – $\zeta = f_r/f_t$, which is the ratio of the area of the actual indicator diagram f_r to the area of the theoretical indicator diagram f_t. Thus, the actual average indicator pressure can be determined by the formula:

$$p_i = \zeta p_i'.$$

For four-stroke marine diesel engines ζ = 0.9...0.96, for two-stroke diesel engines ζ = 0.94...0.98.

In two-stroke engines the value of p', obtained by formula (2.7), is related to the useful stroke of the piston $V_s(\varepsilon - 1)$, where ψ – is the share of compression stroke loss on gas exchange processes:

$$p_i = p_i'(1 - \psi).$$

Values of average indicator pressure p_i, characterising a degree of perfection of a working cycle, for marine low-speed two-stroke engines p_i = 1.8...2.2, MPa, high- and medium-speed four-stroke p_i = 2.5...3.2 MPa.

For preliminary review

An important indicator of any engine is the amount of indicator work produced in one second and called *indicator power* – N_i, kW.

$$N_i = \frac{L_i ni}{60z} = \frac{p_i V_s ni}{60z} = \frac{p_i SFni}{60z},$$

where n – engine speed, min^{-1}; i – number of engine cylinders (if $i = 1$, such power is called cylinder power and is denoted N_{ic}); S – piston stroke, m; F – piston area, m^2; z – coefficient of tactility, for two-stroke engines at which the working stroke occurs at each revolution it is taken equal to – 1, for four-stroke engines – 2.

Indicator efficiency is the ratio of the amount of heat Q_i, converted into work developed by the gases in the engine cylinder L_i (indicator work), to the amount of heat supplied to do this work Q_1:

$$\eta_i = L_i / Q_1.$$

If we take as Q_1 the amount of heat that is released during combustion of a cycle portion of fuel g_{cycl}, its value can be defined as:

$$Q_1 = g_{cycl}\, Q_l,$$

then

$$\eta_i = \frac{Q_i}{Q_1} = \frac{L_i}{g_{cycl} Q_l}.$$

where Q_l is the lower heat of combustion of fuel, kJ/kg.

The work for one hour (3600 sec.) will be $L_i = 3600 N_i$ kJ, and the amount of heat supplied as a result of combustion of fuel of mass G will be equal to GQ_l, kJ.

Indicator efficiency can be found as:

$$\eta_i = \frac{3600 N_i}{G Q_l}. \qquad (2.8)$$

In contrast to the thermal efficiency η_t, the indicator efficiency takes into account heat losses not only caused by heat transfer to the cold source Q_2, but also losses due to heat transfer to the cylinder walls Q_w, as well as from incomplete combustion of fuel and other losses during the working cycle Q_Σ:

$$\eta_i = \frac{Q_1 - Q_2 - Q_w - Q_\Sigma}{Q_1}.$$

The relationship between η_i and η_t is determined by the relative efficiency $\eta_0 = \eta / \eta_{it}$, which takes into account the degree of approximation of the real engine cycle to the ideal one ($\eta_0 = 0.75...0.90$).

Indicator fuel consumption is an important specific indicator that allows estimating the amount of fuel required to produce a unit of indicator power

$$g_i = G/N_i, \text{ kg/(kWh)}.$$

Values of η_i and g_i for marine internal combustion engines are given in Table 2.2.

Table 2.2 – Values of η_i and g_i for main types of marine internal combustion engines

Engine type	η_i	g_i, kg/(kWh)
Low speed engines	0.50...0.57	0.150...0.180
Medium speed engines	0.48...0.55	0.165...0.190
High speed engines	0.35...0.42	0.195...0.255

2.5.2 Effective indicators

Mechanical loss power. Indicator power is, in fact, the power developed by gases in the working cylinder of the engine. During operation, part of this power is spent to overcome friction forces

For preliminary review

in the engine mechanisms, to drive auxiliary mechanisms that ensure its operation, to perform pumping strokes of the piston and others. Power losses associated with the above factors are called the power of *mechanical losses* – N_m. If this power is divided by the working volume of the cylinder, we get a value called the *mechanical loss pressure* $p_m = N_m / V_s$.

Friction losses, as a rule, make up the major part of mechanical losses (55...65%). They are caused by friction in all mating friction pairs, the main ones being friction of piston rings of the piston and cylinder sleeve due to high specific pressure of the rings on the sleeve, as well as friction in crankshaft bearings.

The effective power is the power that can be removed from the engine crankshaft flange – N_e. It is equal to:

$$N_e = N_i - N_m.$$

Mechanical efficiency. The ratio of effective power to indicator power is the *mechanical* efficiency of the engine:

$$\eta_m = N_e / N_i = (N_i - N_m)/N_i.$$

Relative mechanical losses depend on the design, speed and boost level of the engine. Mechanical efficiency characterises the degree of perfection of the engine mechanical design. Its values at rated power mode (usually η_m values are given for 100% engine load mode) are for:

- low-speed two-stroke diesel engines $\eta_m = 0.87...0.94$;
- medium-speed diesel engines $\eta_m = 0.84...0.92$;
- of high-speed diesel engines $\eta_m = 0.75...0.85$.

Average effective pressure is a conditional constant pressure p_e, which during one stroke of the piston would do work equal to the useful work on the engine crankshaft during one operating cycle. It characterises the average specific work of the cylinder per cycle and is one of the most important indicators characterising the engine load, the completeness and dynamics of combustion, the quality of the supercharger and the perfection of the design as a whole. This indicator is used to compare the load boost levels of different engines or different loading modes for the same engine.

The mean effective pressure can be expressed as (Fig. 2.11):

$$p_e = p_i - p_m.$$

If the values of mechanical efficiency and average indicator pressure are known, the value of p_e can be found as:

$$p_e = p_i \eta_m$$

Knowing the average effective pressure p_e it is possible to find the effective power of the engine, which is diverted to external consumers, kW:

$$N_e = \frac{p_e V_s n i}{60 z} = \frac{p_e S F n i}{60 z}.$$

The effective power N_e takes into account the mechanical losses in the engine in addition to the thermal losses, just like p_e:

$$N_e = N_i - N_m = \left(1 - \frac{N_m}{N_i}\right) N_i = N_i \eta_m, \text{ kW}.$$

The values N_e and p_e are the basic energy values of the engine as stated on the engine form.

The effective efficiency η_e characterises the efficiency of the final conversion of thermal energy Q_1 into mechanical work L_e on the engine shaft.

$$\eta_e = L_e / Q_1.$$

If we take, by analogy with the indicator work the effective efficiency:

For preliminary review

$$\eta_i = \frac{3600 N_e}{G Q_l}.$$

Specific effective fuel consumption shows how many kg of fuel must be consumed to produce 1 kW of power at the engine flange for 1 hour:

$$g_e = G/N_e, \text{ kg/(kWh)}, \quad (2.10)$$

then

$$\eta_e = \frac{3600}{g_e Q_l}. \quad (2.11)$$

The g_e and G values refer to the fuel economy of the engine, which is stated on the engine form.

Since $\eta_e / \eta_i = N_e / N_i = \eta_m$ then $\eta_e = \eta_i \eta_m$; $g_e = g_i / \eta_m$. Since $\eta_i = \eta_t / \eta_0$, then $\eta_e = \eta_t \eta_0 \eta_m$.

That is, the effective efficiency η_e is crucial for evaluating the efficiency of the engine as a whole because it is a product:

- thermal efficiency η_t, characterising the degree of perfection of the ideal cycle;
- relative efficiency η_0 characterising the degree of approximation of the design cycle to the ideal cycle;
- mechanical η_m characterising the degree of perfection of engine design and manufacture.

Values of η_e and g_e for marine internal combustion engines are given in Table 2.3.

Table 2.3 – Values of η_e and g_e for main types of marine internal combustion engines

Engine type	η_e	g_e, kg/(kWh)
Low speed engines	0.42...0.54	0.160...0.185
Medium speed engines	0.40...0.52	0.165...0.215
High speed engines	0.32...0.41	0.205...0.265

2.6 New approaches in organisation of operating processes in marine four-stroke engines. Miller cycle

Tougher requirements to fuel efficiency and environmental safety of marine power plants have led to the fact that many engine-building companies are reconsidering their approaches not only to the design of newly created engines, but also to the organisation of their operating process.

As one of the most promising directions, the use of the Miller cycle in high-force marine medium-speed four-stroke diesel engines is currently considered. The ideal Miller cycle for diesel engines is shown in Fig. 2.17. Its use allows simultaneously increasing the fuel efficiency of the engine due to the extended expansion and reducing the formation of nitrogen oxides in the exhaust gases by reducing the heat intensity of the operating process.

The extended expansion duty cycle was patented in 1947 by American engineer Ralph Miller. The basic idea behind the Miller cycle is to reduce the degree of compression and increase the degree of expansion of the working medium. A similar idea was expressed in the second half of the 19th century by the British engineer James Actinson*. To realise the proposed principle, Actinson developed special mechanisms for driving the working piston, which were not widely spread due to their complexity. Miller managed to realise Actinson's ideas in engines with a traditional crank mechanism.

In his patent, Miller considers two variants for organising said operating process with too early or too late closing of the inlet valve.

The theoretical cycles for both ways of realisation of the considered operating processe are presented in Fig. 2.18.

In the first case, called «shortened compression», as the piston moves from the BDC until the inlet valve closes, some charge is displaced into the intake manifold, and only that part of the charge remaining in the cylinder is compressed (Fig. 2.18 *a*).

For preliminary review

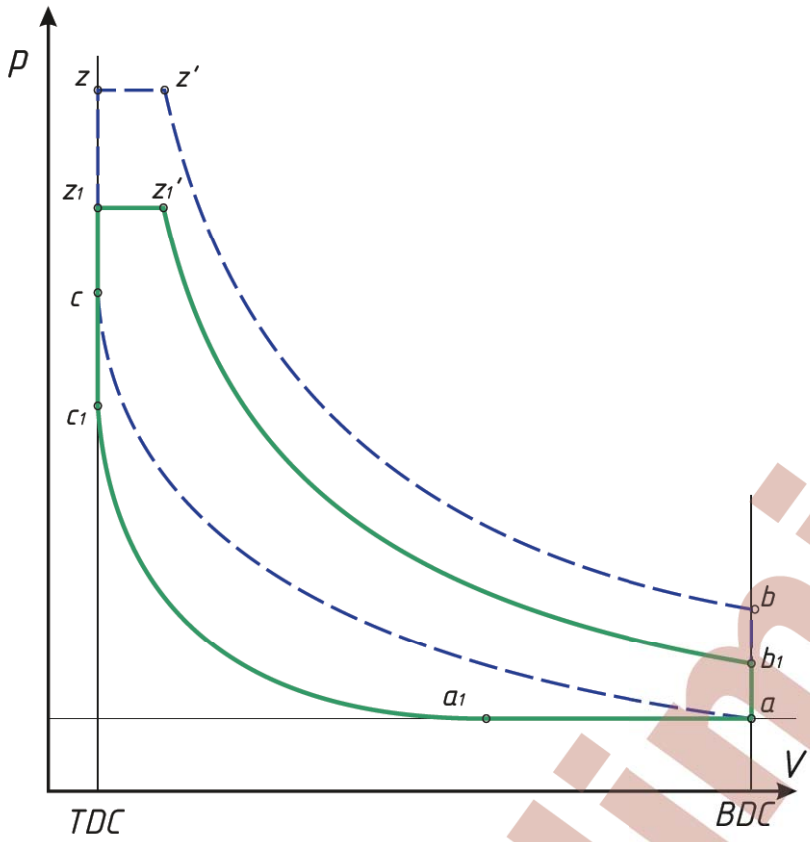


Figure 2.17 – Comparison of ideal cycles of diesel engines operating on the classical (a, c, z, z', b, a) cycle and the Miller cycle (a, a_1, c_1, z_1, z'_1, b_1, a) (adapted from [3])

In the second case, called «shortened intake», the valve closes long before the piston reaches the BDC (Fig. 2.18 *b*). In this case, from the moment of valve closing until the piston arrives at the BDC, the charge is pre-expanded and only then compression follows.

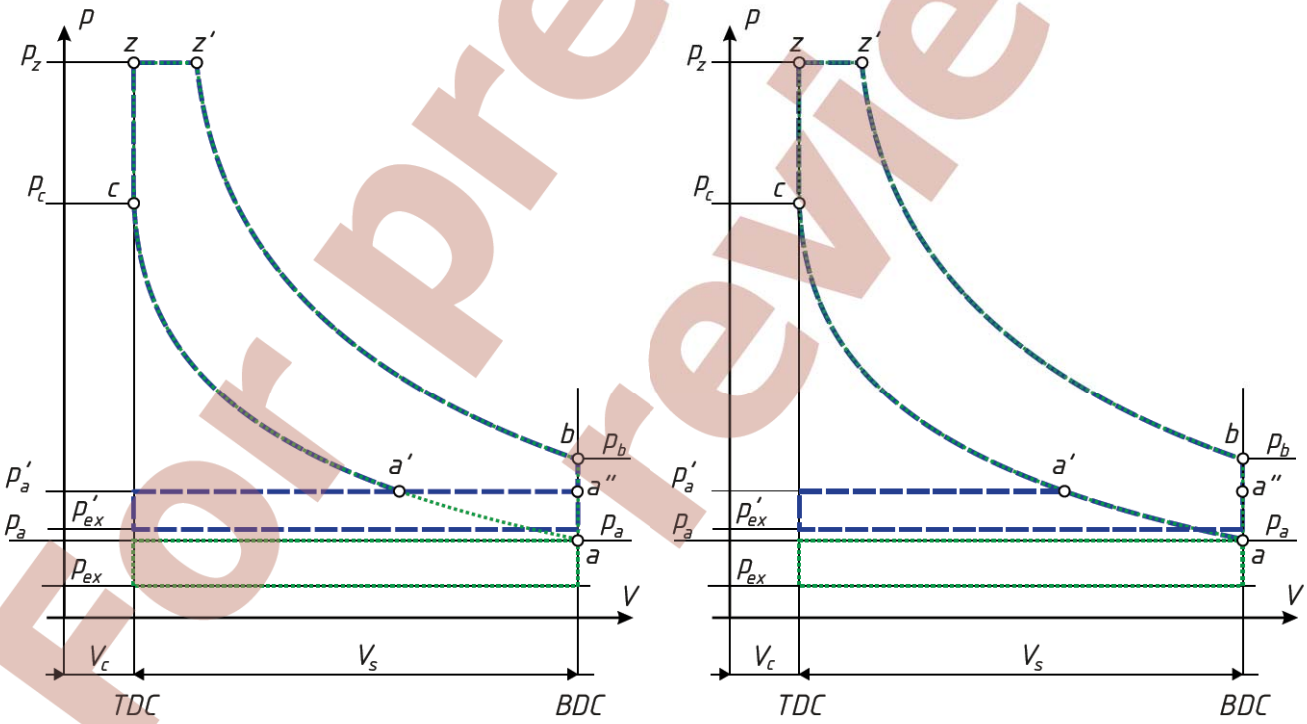


Figure 2.18 – Variants of Miller cycle realisation: a – with shortened compression; b – with shortened inlet. – – – – Miller cycle; ····· – basic cycle (adapted from [3])

Ultimately, both variants lead to a decrease in the actual compression ratio of the working mixture relative to the geometric one. In the theoretical cycle (Fig. 2.18), compression starts not at point a, as in the basic cycle, but at point a'. At the same time, the degree of expansion remains un-

For preliminary review

changed, resulting in an increase in the efficiency of the operating process.

The main disadvantage of both variants of Miller cycle realisation is that, all other things being equal, the fill factor of the working cylinder decreases, which means that the power removed from the engine is reduced.

It is possible to compensate for the loss of power when switching to Miller cycle by increasing the initial air pressure using high supercharging. Fig. 2.18 shows the basic engine cycle with Miller cycle variants superimposed on it. The figure shows that to obtain Miller cycle parameters identical to the basic cycle, the pressure at the inlet to the cylinder must be increased from p_a to p'_a.

The only difference between theoretical cycles with shortened compression and shortened expansion is the section between the onset of compression a' and BDC.

At shortened inlet, in the cycle section a'-a-a', with the valves closed, there is a preliminary expansion of the working medium (section a'-a) followed by its preliminary compression (section a-a'). Thus, for this method of realisation of the Miller cycle, the compression process consists of two stages, preliminary and main. The moment when the main compression starts is understood as the moment when the pressure in the working cylinder exceeds the pressure in the inlet receiver.

In the case of shortened compression, at cycle section a'-a''-a', the charge is injected into the cylinder (section a'-a'') and then, through the open valves, part of the charge is forced back into the intake receiver by the moving piston (section a''-a'). In this case, the theoretical moment of the beginning of compression coincides with the closing of the inlet valves.

In theoretical cycles, both sections are characterised by zero work and all other sections of the cycle are identical.

Considering the high level of boosting of modern marine diesel engines, it is particularly important that the use of the Miller cycle allows to reduce the level of thermal stress of the operating process. This is explained by the fact that in order to realise the Miller cycle with the same maximum pressure as in the basic cycle, air must be supplied to the engine with a higher initial pressure, which is achieved by using higher boost pressures. Given that the air is subjected to intermediate cooling after the turbocharger, it enters the engine cylinders at a lower temperature than that which, at the same pressure, corresponds to the compression process at point a' of the base cycle. As a result, the temperature values at all key points of the cycle decrease, which is clearly seen from the diagram in *TS*-coordinates shown in Fig. 2.19 *a*.

In the theoretical cycle with shortened inlet, the working medium is pre-expanded and therefore its temperature at the start of pre-compression is lower. During the subsequent compression, the charge temperature increases and by the time the main compression starts, it becomes equal to the air temperature in the air receiver. Since the piston has already travelled part of the stroke, the temperature at the end of compression, all other things being equal, will be lower than in the basic cycle. Correspondingly, the temperature values at the other key points of the cycle also decrease. The latter can be clearly seen from the diagram of the operating process in *TS*-coordinates presented in Fig. 2.19 *a*.

A similar decrease in temperatures at all key points is observed in the Miller cycle with shortened compression. In this case, the charge temperature in the cylinder remains constant, equal to the air temperature in the inlet receiver during the entire piston stroke from BDC to the inlet valve closure (section a'-a''). Isoentropic (adiabatic) compression, as in the first case, starts at point a′ with the same initial parameters of the working medium state as in the case of the process with shortened inlet.

The character of the pressure curve on the *pV*-diagram is directly related to the work done by the gases during the cycle. The temperature curve on the *TS*-diagram does not directly characterise the work, but the level of operating temperatures affects the value of heat losses, which ultimately affects the cycle efficiency (Fig. 2.19). Indirectly, the change in the Miller cycle gas exchange parameters also affects the quality of fuel combustion, as the temperature decrease leads to a change in the character of pre-flame processes in the engine combustion chamber.

Figure 2.19 *b* shows the combined diagrams of the Miller cycle operating processe and the base engine operating processe in *pV*-coordinates, similar to those shown In Fig. 2.19 *a* in

For preliminary review!

TS-coordinates. The figures show that at the same pressures at the key points of the cycle lower temperatures operate at the same pressures, which indicates a decrease in the thermal stress of the operating process.

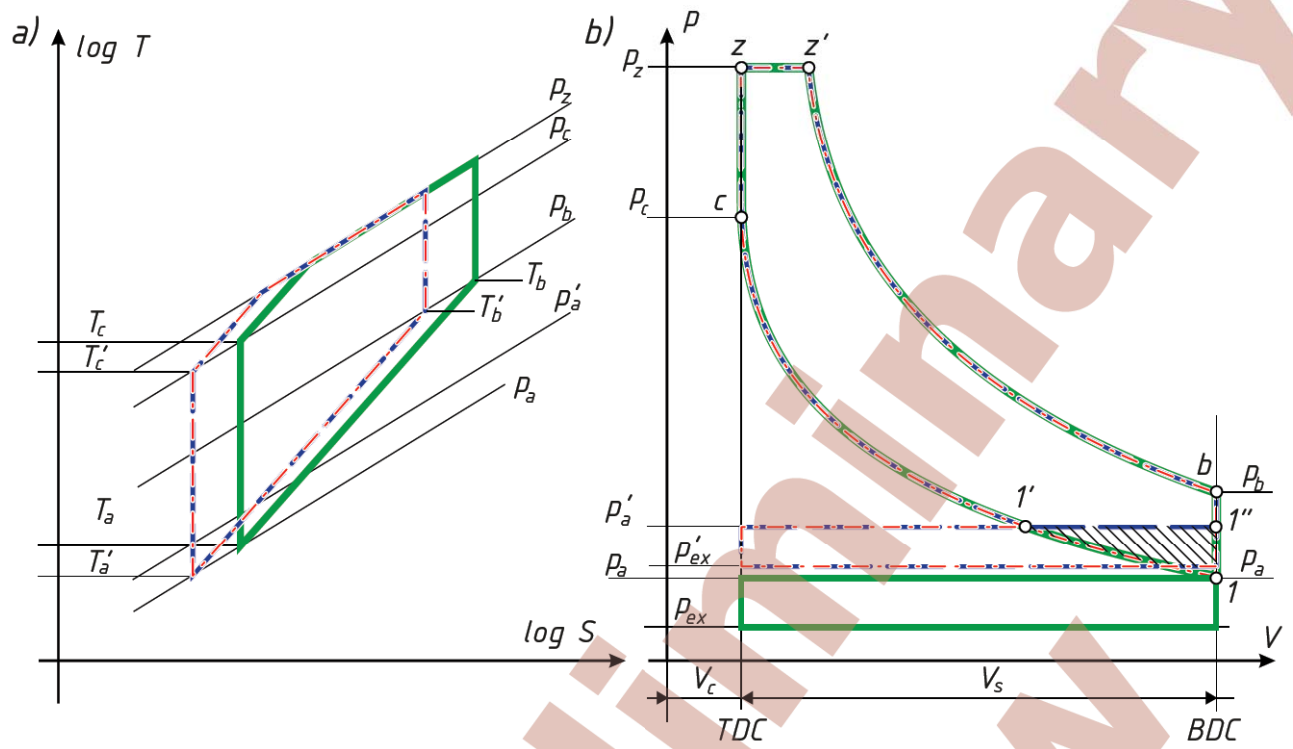


Figure 2.19 – Combined diagrams of operating processes of the basic engine and the Miller cycle with shortened compression and intake in *TS*-coordinates (*a*) and in *pV*-coordinates (*b*): ——— – basic cycle; – – – – – – with shortened compression; – – – – – – with shortened intake (adapted from [3, 4])

If we analyse theoretical cycles of engines operating under the Miller cycle, we can conclude that, all other things being equal, their efficiency will be lower than in the basic cycle due to the loss of part of positive work in the gas exchange section between the compression line and BDC (shaded area in Fig. 2.19 *b*). The loss of part of useful work in the gas exchange section, called «Miller losses», can be as high as 0.5 per cent, all other things being equal. In real cycles «Miller losses» are compensated, due to higher supercharging pressure and lower heat losses associated with reduction of the overall heat intensity of the operating process.

While maintaining boost pressure, switching to the Miller cycle results in lower pressures at key points in the cycle, resulting in lower mechanical stress, which in turn results in higher mechanical efficiency of the engine.

In order to quantify the influence of the selected gas distribution phases on the character of the operating process, a dimensionless indicator called the Miller coefficient is used:

$$m = S_M/S$$

where S_M is the part of the piston stroke in the Miller cycle spent on compression; S is the full stroke of the piston.

On this basis, in Miller cycle engines it is necessary to separate the concepts of compression ratio and expansion ratio. The expansion ratio is determined from the relationship $\varepsilon = (V_c + V_s)/V_c$ (Fig. 2.19 *b*), while the compression ratio depends on the Miller ratio and is determined from the relationship $\varepsilon_M = m\,(\varepsilon - 1) + 1$.

In the most general form, the expression can be used to assess the influence of the Miller coefficient on the efficiency of the operating process, all other things being equal:

$$\Delta\eta = \Delta\eta_{cce} + \Delta\eta_{egp},$$

where $\Delta\eta_{cce}$ – is the closed loop efficiency; $\Delta\eta_{egp}$ – efficiency of the gas exchange process.

Figure 2.20 shows the effect of operating processe efficiency components on the overall cycle

For preliminary review

efficiency as a function of the Miller coefficient.

The figure shows that the efficiency of the closed cycle increases almost directly proportional to the decrease of the Miller coefficient. The efficiency of the gas exchange process remains practically unchanged within the range of change $m \approx 0.85...1.0$. Further increase of m is accompanied by a sharp increase in throttling losses and «Miller losses». For the majority of engines the optimum value of m lies within the range 0.68...0.72.

Based on the above, it can be concluded that the use of the Miller cycle allows to obtain a number of advantages in the organisation of the operating process of engines without increasing their thermal and mechanical stress.

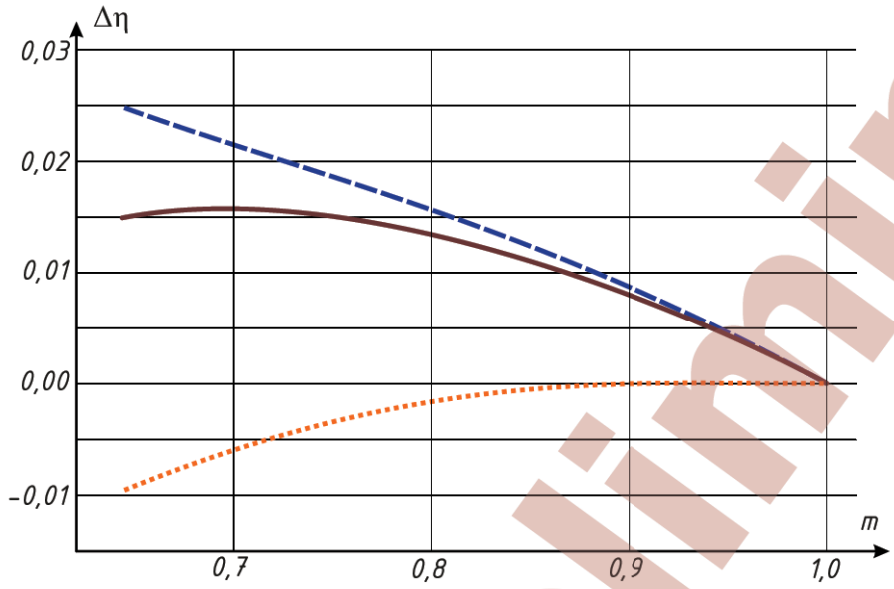


Figure 2.20 – Operating processe efficiency (———) as the sum of closed-loop efficiency (– – – –) and gas exchange process efficiency (·····)

Some of the variants of operating processe organisation using the Miller cycle are shown in Fig. 2.21. For comparison in all figures the basic engine cycle is shown with a dashed line.

Fig. 2.21 *a*, shows the Miller cycle with initial pressure corresponding to the basic operating process. It can be seen that the application of this cycle allows to reduce the maximum pressure and the pressure at the end of expansion, which leads to a decrease in mechanical losses in the engine and an increase in its thermodynamic efficiency.

The disadvantage of the variant considered in Fig. 2.21 *a* is the power reduction caused by the reduction of the cylinder weight charge. This disadvantage can be eliminated by increasing the air density by increasing its initial pressure. In this case, the values of parameters of the working medium state at key points will be the same as in the basic cycle (Fig. 2.21 *b*). At unchanged cyclic fuel supply, this allows increasing the excess air ratio, which leads to improved combustion conditions.

The loss of power can be compensated by increasing the geometric compression ratio (Fig. 2.21 *c*). If the initial pressure is equal, the efficiency of the operating process will be higher compared to the basic cycle. However, the low pressure at the end of expansion may not be sufficient for efficient operation of the turbocharger. This disadvantage can be overcome by increasing the proportion of heat input at the expansion line (Fig. 2.21 *d*), which can be achieved by increasing the cycle supply or by shifting the combustion process to the expansion line, by feeding the fuel later. In addition, such an approach would increase engine power. This approach can be even more effective when combined with an increase in compression ratio (Fig. 2.21 *e*).

An integrated approach to the organisation of the Miller cycle operating process can be considered as a promising one, when high supercharging, increased compression ratio and transfer of most of the heat input to the expansion line are used simultaneously. This maximises the use of cylinder space to produce work without increasing mechanical and thermal stress. The disadvantage of such a operating processe is the higher pressure at the end of expansion (Fig. 2.21 *f*). However, with highly supercharged systems, especially with two stages of compression, this potential may just be necessary for efficient turbocharger operation.

For preliminary review

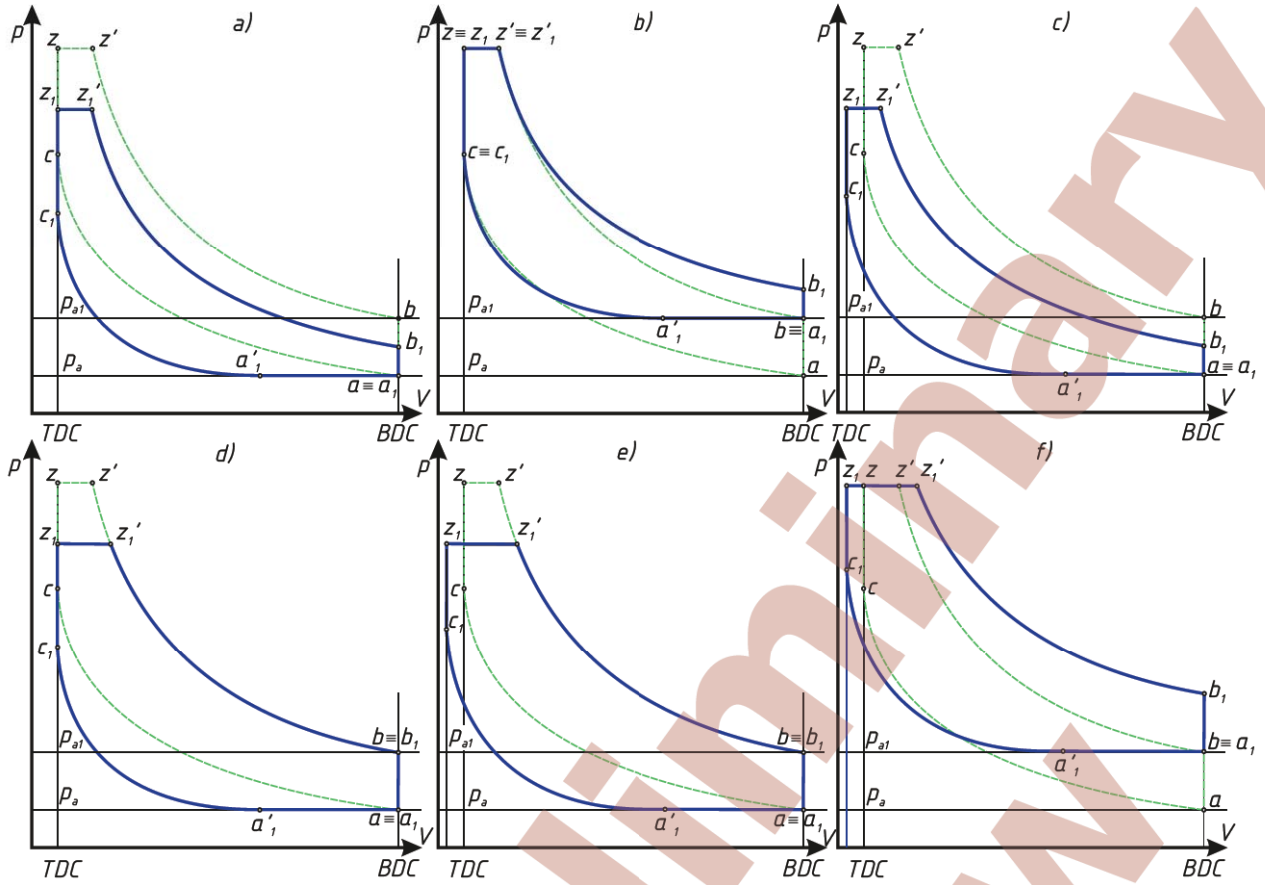


Figure 2.21 – Variants of Miller cycle implementation in comparison with the basic cycle: *a* – with the same initial pressure; *b* – with increased initial pressure and the same maximum pressure; *c* – with the same initial pressure and increased compression ratio; *d* – with the same initial and final pressure due to extended heat input; *e* – with increased compression ratio and the same final pressure; *f* – with the same initial and maximum pressure, increased compression ratio, extended heat input (adapted from [5])

Various combinations of the above mentioned variants allow to expand considerably the possibilities of the organisation of the operating process of a marine diesel engine depending on what tasks are set as priorities in its design.

The character of gas exchange in real engines operating under the Miller cycle with shortened intake and shortened compression is shown in Fig. 2.22. For comparison, each of the figures shows the process of gas exchange of the same engine when it operates under the traditional cycle. In this case, the power of the engine operating under the Miller cycle, due to increase of supercharging pressure (p'_s), is the same as that of the engine with the traditional gas exchange system having a lower supercharging pressure (p_s).

As already mentioned, in a theoretical cycle, late or early closing of the intake valves leads to the same result. In real cycles, there are some differences:

- with shortened intake, the valves start to close very early; therefore, the difference between cylinder pressure and boost pressure results in high throttling losses, which in turn contributes to additional charge cooling (section BDC-3″ Fig. 2.22*b*). Pre-expansion prior to compression leads to a lowering of the working medium temperature directly in the working cylinder, which is why the Miller cycle with shortened intake is also called a cycle with internal charge cooling;
- in case of late closing of the inlet valve and release of part of the air charge, there are also some throttling losses, which leads to some pressure rise in the section of the charge ejection stroke (section BDC-3‴ Fig. 2.22*b*).

With shortened compression, heat transfer must also be taken into account: the air charge forced out of the cylinder by the piston has already been heated, and this heat accumulates in the intake duct until the intake valve is opened again in the next cycle. This partially lowers the theoretical cooling of the cylinder charge, compared to early closing (Fig. 2.23).

For preliminary review

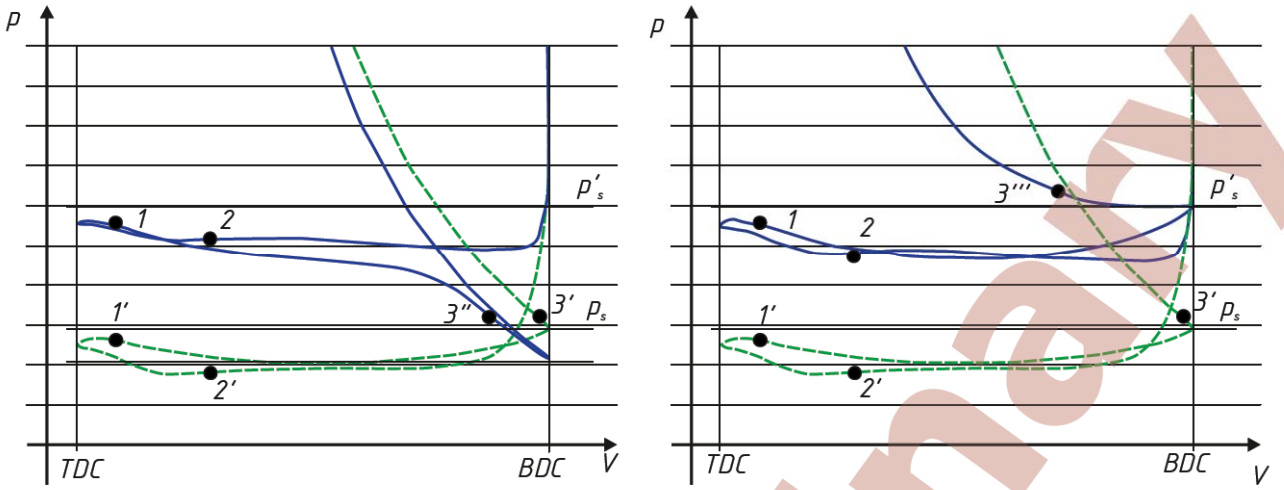


Figure 2.22 – Gas exchange in engines operating on Miller cycle: *a* – with shortened intake; *b* – with shortened compression. 1, 1′ – opening of the exhaust valve; 2, 2′ – closing of the exhaust valve; 3′ – closing of the intake valve in the basic cycle; 3″ – early closing of the intake valve; 3‴ – late inlet valve closing (adapted from [4])

Thus, the use of shortened compression allows the maximum pressure and cycle temperature to be reduced more effectively. The use of shortened compression is particularly effective in medium-speed engines operating at constant speed. Relatively low speeds of rotation of the ICE allow to provide an optimal law of motion of the intake valve, providing the maximum passage cross-section of the valve slit.

As the speed increases, the shortening of the opening and closing times of the valves imposes limitations due to mechanical loads in the valve actuator. This often results in the need to reduce valve travel, which increases throttling losses. In addition, at lower speeds and consequently lower boost pressures, closing the intake valve too early leads to a significant deterioration in cylinder filling.

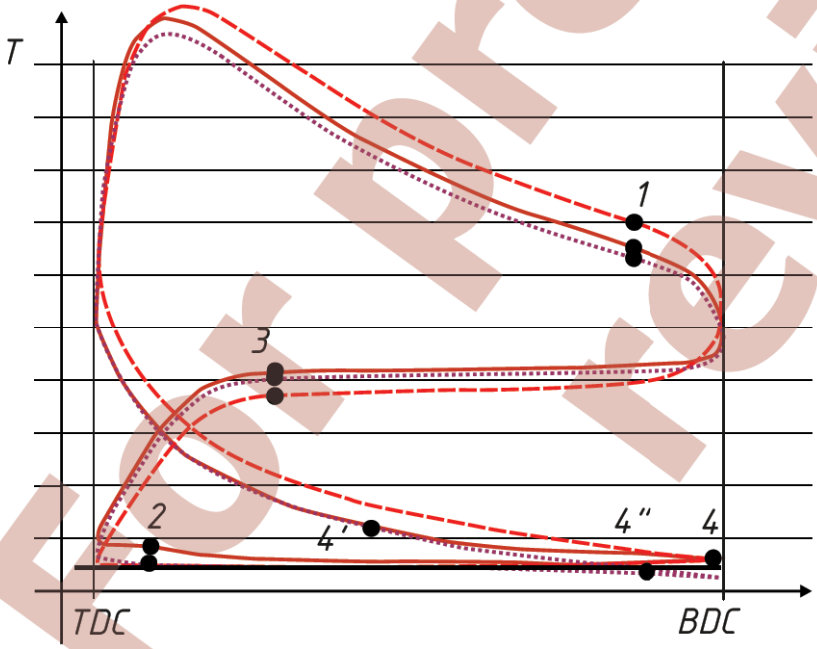


Figure 2.23 – Temperature change in the operating process for the basic cycle (– – –), in the cycle with shortened compression (——) and in the cycle with shortened inlet (- – - -). 1 – exhaust valve opening torque; 2 – exhaust valve closing torque; 3 – inlet valve opening torque; 4 – inlet valve closing torque in the basic cycle; 4′ – inlet valve closing torque in the shortened compression cycle; 4″ – inlet valve closing torque in the shortened inlet cycle (adapted from [4])

Therefore, for high-speed engines and engines operating in a wide range of load-speed modes, the Miller cycle with shortened compression is preferred, and for engines operating at a constant speed at modes close to rated power predominantly use the Miller cycle with shortened intake.

The use of Miller cycles in modern marine four-stroke engines over the previous two decades has enabled the majority of leading engine-building companies to successfully solve a set of problemsin converting medium- and high-speed engines to run on heavy fuels. In addition to significant

For Preliminary review

changes in the design of such engines, which will be discussed further in the relevant chapters, their operating process has undergone radical changes. This is primarily due to the fact that heavy fuels are characterised by a sufficiently high activation energy, which means the amount of heat that must be supplied to the fuel before it reacts with oxygen in the air.

Taking into account that the time for fuel combustion in medium- and high-speed engines is an order of magnitude shorter than in low-speed engines (in LSE at 100 min^{-1} 80...100 ms are allocated for the combustion process, in LSE at 1000 min^{-1} 5...8 ms), for qualitative combustion of fuel the activation energy should be supplied as soon as possible. The transition from heat supply to fuel to heat release as a result of reactions with air oxygen is called the activation threshold of the combustion process, and all processes preceding combustion are called pre-flame. To increase the rate of heat supply to the fuel, in order to reduce the time of pre-flame processes, it is possible to increase the temperature of the air charge and reduce the size of the fuel aerosol droplets. In practice, the former is achieved by increasing the temperature at the end of compression, and the latter by increasing the fuel atomisation pressure. To increase the temperature at the end of T_c compression to 900...1200 K, in this kind of engines, the compression ratio and boost pressure are increased. As a result, the pressure at the end of compression p_c, can reach 12...17.5 MPa, which is on the boundary close to the maximum allowable pressures based on the strength limits of the materials used. In this connection combustion in the engine working space should take place without significant pressure increase.

This is achieved by shifting the combustion process from the region close to the TDC to the expansion line. Theoretically, such a cycle is closer to the ideal Diesel cycle than to the Trinkler-Sabathé cycle. The shift of the combustion process to the expansion line is achieved by reducing the injection advance angle, as well as by using special fuel supply laws, including staged multiple injection.

Fig. 2.24 shows the pressure change as a function of the rotation angle in a conventional diesel engine operating process and in the Miller cycle with the combustion process shifted to the expansion line. The degree of pressure rise λ in such engines usually lies within 1...1.3. To achieve the specified parameters of the working medium by the end of compression, the boost pressure is increased to 0.3...0.4 MPa and the compression ratio to 13...16.

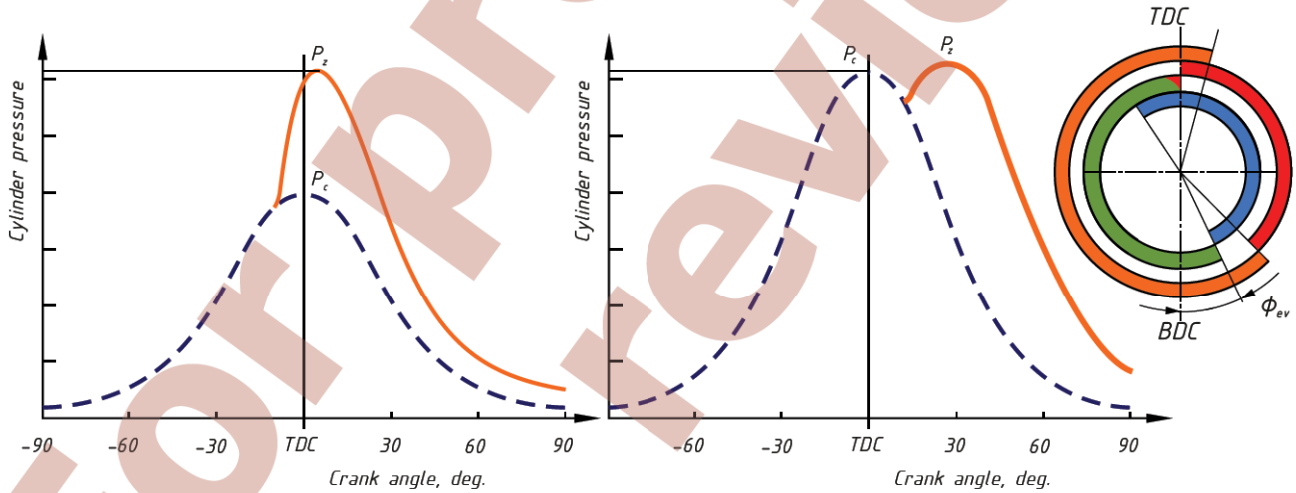


Figure 2.24 – Pressure variation in the operating processe as a function of crank angle. *a* – conventional operating processe; *b* – Miller process. – – – – compression line; ——— combustion line; *c* – gas distribution phase diagram for Miller process

Decrease in thermal efficiency of the piston part of the engine associated with an increase in the degree of pre-expansion ρ is compensated by the use of turbochargers with a higher efficiency capable of effectively working with a higher temperature difference.

The activation energy of the combustion process is the amount of heat that must be supplied to the fuel to initiate reactions between the fuel molecules and the oxygen in the air.

For preliminary review

2.7 Classification of marine internal combustion engines

Modern marine diesel internal combustion engines can be classified according to the following main characteristic features.

By the way the working cycle is carried out:

- four-strokes, where the operating cycle is completed in two crankshaft revolutions;
- two-stroke, where the operating cycle *is* completed in one revolution of the crankshaft.

By purpose:

- main reversible with direct power transmission to the propeller;
- non-reversible main units operating in conjunction with a reversing gearbox or reversing coupling;
- main non-reversible, powered by electric generators or adjustable pitch propeller;
- auxiliary non-reversible, operating to drive auxiliary mechanisms (diesel generators, diesel compressors, etc.).

By the method of filling the cylinder:

- unsupercharged engines with air intake by the piston (in four-stroke engines) or cylinder cleaning by air supplied by the purge pump (in two-stroke engines);
- supercharged cylinders are filled with pressurised air generated by a mechanical or gas turbine driven supercharger.

By the type of fuel used:

- engines operating on liquid fuels, which in their turn are subdivided into engines operating on light (diesel fuel) and heavy (engine, gas turbine and fuel oil) fuels. Engines, which without design changes can operate on liquid fuels of different fractional compositions, are called multi-fuel engines;
- gas-diesel engines are predominantly fuelled by gaseous fuel, and liquid fuel is fed into the combustion chamber in small quantities only to activate the combustion process. As a rule, if necessary, these engines can be converted to liquid fuel;
- gas engines designed to run solely on gas fuel (usually operating on the Otto cycle).

By the method of ignition of the fuel-air mixture:

- with compression auto-ignition (diesel engines);
- with pilot ignition (gas-diesel engines);
- with forced ignition, from an electric spark or other source of open flame (Otto cycle gas engines). In ship power engineering such engines are used on gas carriers.

By method of mixing:

- with internal mixing, air and fuel enter the cylinder separately and are mixed inside the cylinder (diesel engine);
- external mixing, i.e. the cylinder is supplied with a ready combustible mixture of fuel and air, which is formed in a special mixer (gas-diesel and gas engines).

By cylinder arrangement:

- single-row with vertical cylinders in one plane;
- with angled cylinder rows (V-shaped);
- single or double row with counter-rotating pistons.

By S/D ratio:

- engines with $S/D \leq 1$ are called short stroke engines;
- engines with $S/D > 1$ are called long stroke engines.

By design of combustion chambers:

- with non-separated single-cavity combustion chambers (mainly medium and larger capacity diesels);
- with semi-separated combustion chambers (diesels with combustion chambers in the cap and piston);
- with separated combustion chambers with two or more cavities (pre-chamber and swirl-chamber).

For preliminary review

By constructional design of the crank mechanism:

- crossheads, in which the lateral (normal) forces are absorbed by the crosshead slider moving along the parallel. The piston itself is only subjected to forces along the cylinder axis;
- trunk, in which lateral (normal) forces are absorbed by the developed lower part of the piston (trunk), which transmits them to the cylinder walls.

In the direction of rotation of the crankshaft:

- having one constant direction of crankshaft rotation left or right;
- reversible, where the direction of rotation of the shaft can be reversed.

By crankshaft speed:

- low-speed engines (n = up to 350 min^{-1});
- medium-speed engines (n=350...750 min^{-1});
- high-speed engines (n=750...2500 min^{-1}).

By average piston velocity.

Engines are divided into 3 groups according to the average piston speed (c_m):

- c_m < 6.5 m/s – with low piston speed;
- c_m (6.5...9) m/s – with medium piston speed;
- c_m > 9 m/s – with high piston speed.

Diesel engine labelling. In order to distinguish between individual engine designs, they are labelled. Each manufacturer of marine diesel engines has its own engine labelling system.

It should be noted that companies do not always follow the declared system of designations exactly. In order to decipher various factory markings, it is necessary to be guided by the relevant documentation supplied by the manufacturer together with the engine. Further, for example, let us consider the current labelling system for medium, high (Fig. 2.25) and low-speed marine engines of MAN (Fig. 2.26), which occupies a significant part of the marine engine market.

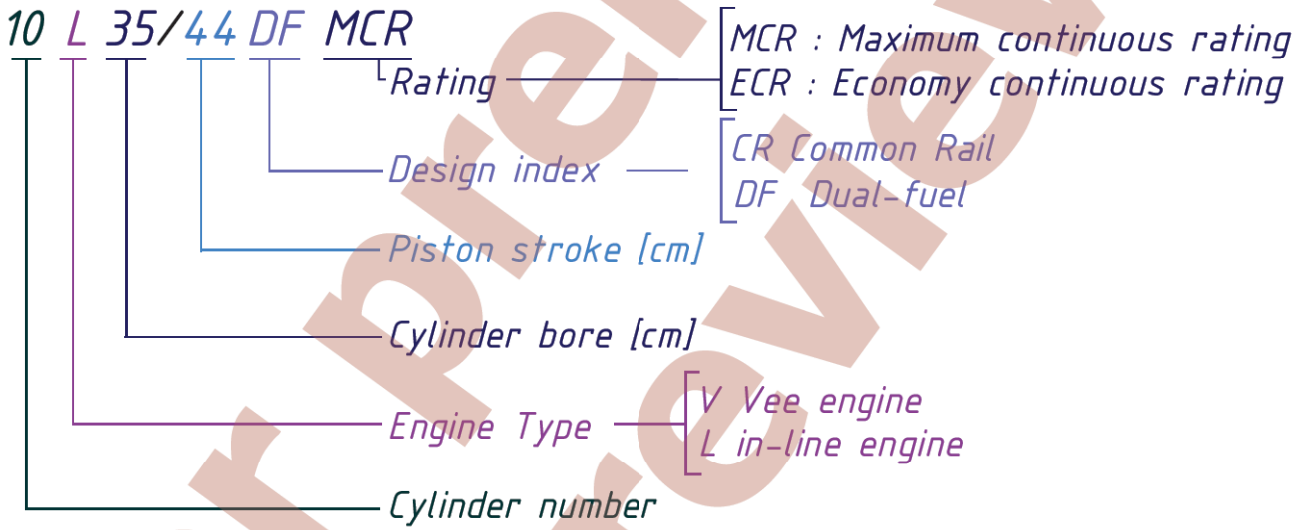


Figure 2.25 – Designation of MAN medium and high speed engines [6]

For the LSE model range, MAN has introduced a designation system based on the piston stroke level, the principle of which is shown in Fig. 2.26.

Ultra-long stroke engines are diesel engines with S/D=(3.54...4.4), long stroke engines with S/D=(2.8...3.24), short stroke engines with S/D=(2.45...2.88).

Nowadays, another class of engines with even higher S/D=(4.4...5.0) ratio has been added to the existing sizes, which are labelled with G index. Such level of long stroke allows reducing nominal diesel engine revolutions to 58...68 min^{-1}. As a result, the propulsive efficiency of the entire power plant increases significantly, which leads to a 4...9% reduction in fuel consumption compared to other types of engines.

For preliminary review

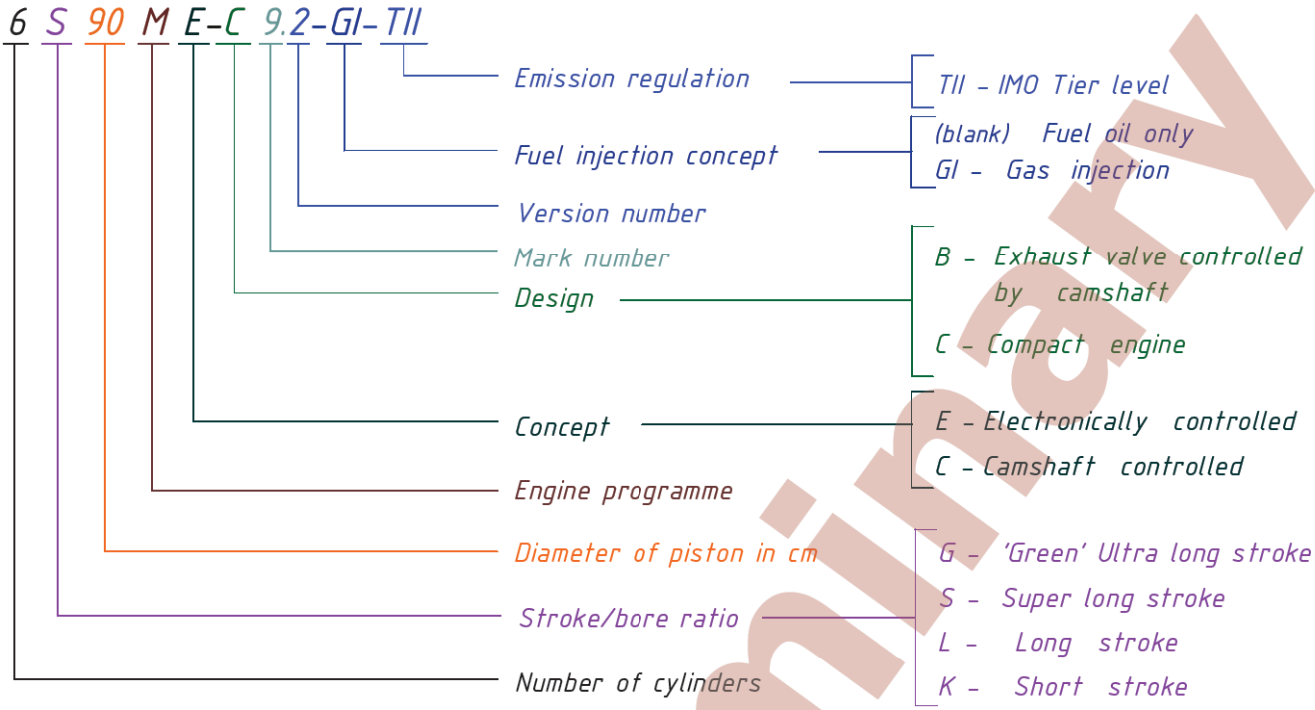


Figure 2.26 – Designation of MAN low-speed engines [7]

2.8 Architecture of marine internal combustion engines

The architecture of an internal combustion engine is understood as a set of layout solutions that are determined by the choice of the basic engine design scheme.

When selecting layout solutions, designers proceed from a number of requirements, which primarily depend on the peculiarities of marine engine operation. The main requirements include:

- the ability of the engine to maintain its serviceability in a wide range of load and speed modes for a specified, sufficiently long-time intervals, since loss of engine serviceability in most cases leads to a vessel accident;
- high engine life, which means the longest possible service life during which the diesel engine must operate reliably and economically until overhaul;
- as simple and technologically advanced a engine design as possible to facilitate its manufacture and installation;
- convenience for carrying out maintenance and repair work on the engine, as well as ease of monitoring the condition of its structural elements during operation;
- accessibility for observation and inspection of all the most critical engine components and parts (bearings, crank mechanism parts, etc.);
- restrictions on dimensions, weight, installed power and layout of the ship's power plant.
- maximum safety of engine maintenance under all operating conditions, observance of all safety rules stipulated for engines. Depending on what requirements to the engine are prioritised, one or another layout scheme is preferred, but regardless of this, the requirement for high reliability of the marine engine is always in the first place.

Of all the variety of architectural forms, only a few layout schemes have found application in ship power engineering, the main ones of which are shown in Fig. 2.27, some of these schemes are peculiar only to marine engines.

Trunk engines (Fig. 2.27 *a*, *b*) are characterised by simple construction and good mass-size indicators. The disadvantages of these engines include increased wear of the working cylinder sleeve due to pressing the piston against the sleeve wall under the action of the normal component of the force directed along the connecting rod and the inability to separate the piston cavity from the crankcase cavity. The latter leads to the fact that into the crankcase gases burst from the piston space, deteriorating the quality of lubricating oil and causing various kinds of deposits on the inter-

For preliminary review

nal surfaces of the engine.

As a result, all other things being equal, engines of this type have a shorter service life than crosshead engines. However, in newly developed diesel engines, a number of design solutions, which will be discussed below, allowed to bring the life of tronic engines to the same order of magnitude as low-speed crosshead engines.

The most characteristic layout of the trunk engine is an in-line arrangement of cylinders (Fig. 2.27 *a*). Single-row engines are simple in design and sufficiently technological in manufacture. These advantages, as well as a great experience of creation and operation of engines with vertically arranged cylinders cause their wide application in ship power engineering. Such arrangement allows to simplify enough the maintenance and repair of engines in the engine room. In particular, removal of heads, removal of pistons, etc. can be carried out using standard overhead hoisting and transporting mechanisms without additional accessories. The location of the various units of the engine systems on the sides of the crankcase makes it quite easy to monitor them during operation, service and repair. In-line engines are usually produced with 3 to 9 cylinders.

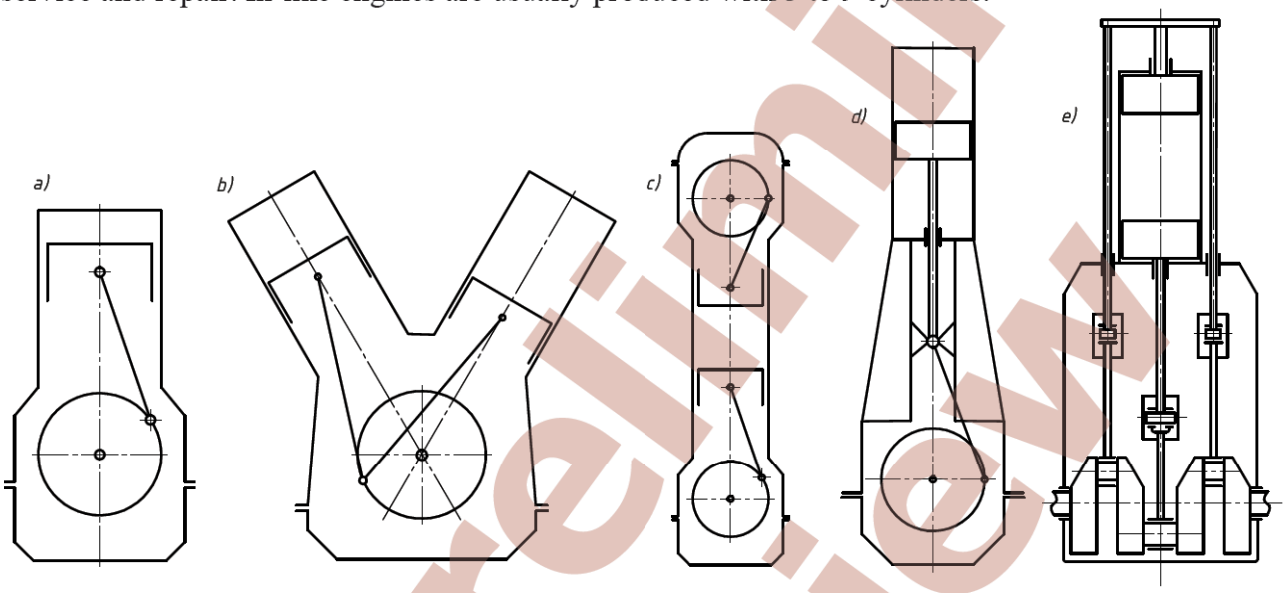


Figure 2.27 – Architectural forms of marine internal combustion engines: *a* – trunk engines in-line; *b* – trunk engines V-shaped; *c* – trunk engines with oppositely moving pistons; *d* – crosshead piston engines; *e* – crosshead engines with oppositely moving pistons

If it is necessary to increase the specific power, a two-row arrangement of working cylinders is used (Fig. 2.27 *b*), placing them relative to each other at an angle of 30...120°. This arrangement of cylinders is called V-shaped (after the shape of the Latin letter V). It allows, practically at the same power to halve the length of the engine and increase the rigidity of such critical structural elements as the frame and crankshaft. The angle between the cylinder axes, called the camber angle, is determined by the purpose of the engine, requirements to the size and order of operation of the cylinders located in one row. The disadvantage of this scheme is a more complex design of CRM and some inconvenience in maintenance, especially in the cramped conditions of the engine-boiler room of the ship. marine engines of this layout usually have from 6 to 24 cylinders and more.

Both two-stroke and four-stroke operating process is realised in trunk engines, but four-stroke trunk engines are predominantly used in the fleet. Relatively small mass of moving parts allows to force the engines of this type by speed without excessive increase of dynamic loads on bearings. In this connection, mainly medium and high-sided engines are built according to this scheme. At the same time, some Asian manufacturers, such as Akasaka Diesels, Hanshin EL, Makita, Matsui Iron Works, Niigata Engineering offer a range of low-speed trunk diesel engines. In particular, the Japanese company Hanshin EL produces six-cylinder diesels with cylinder diameters from 22 to 58 cm. The largest engine of this series, model LF58A, with a cylinder diameter of 580 mm and a piston stroke of 1050 mm develops power of 770 kW/cyl. at a speed of 190 min^{-1}.

For preliminary review

Engines with oppositely moving pistons of trunk type. Out of a rather extensive variety of layout schemes of trunk engines with oppositely moving pistons, the layout shown in Fig. 2.27 *c* has gained some popularity in the fleet. The development of this layout is associated with the name of the outstanding German engineer Hugo Junkers*.

Engines of this layout are built only as two-stroke engines, as it is quite simple to realise straight-through purging in them by placing the function of gas flow control on the pistons. In these engines, one of the cylinder pistons controls the intake and the other – the exhaust, for this purpose the cranks of each cylinder are displaced one relative to the other by an angle of 10...15°.

The advantage of these engines is the complete absence of a timing mechanism and gas joint between cylinders and cylinder heads.

The disadvantages are that the piston controlling the exhaust is washed with exhaust gases and operates under particularly difficult conditions, in addition, more power is removed from the crankshaft controlling the exhaust and this shaft is more loaded.

Currently, the only supplier of such engines for the fleet is the American company Fairbanks Morse, which produces engines model 38TD8-1/8 in 6, 9 and 12 cylinder version with a cylinder diameter of 206.4 mm and piston stroke of 254 mm. These engines have a cylinder power of about 260 kW at a speed of 900 min^{-1}. Engines of similar layout of 2D100 and 10D100 type (10D 20.7/(2×25.4)) were produced in the USSR and post-Soviet countries, they were used on marines as part of diesel generators.

Crosshead engines (Fig. 2.27*d*). This engine layout has become quite widespread in the Navy, having been transferred to internal combustion engines from steam engines, on which it was quite well developed. This is primarily due to a number of significant advantages, which include:

- in this type of engines only forces directed along the cylinder axis are perceived by the piston, which significantly reduces wear of the cylinder sleeve guide surface, piston and piston rings. Lateral (normal) components of forces acting in the CRM are perceived by the crosshead sliders and are transferred to the parallels, which are placed in the internal (crankcase) space of the engine. This allows organising lubrication of rubbing surfaces under pressure, as a result of which mechanical losses in the CRM are reduced;

- the rod connection of the piston and crosshead allows to easily separating the engine crankcase cavity from the piston space. This makes it possible to organise separate lubrication of parts located in the crankcase (crankshaft bearings, crosshead assembly, camshaft bearings, etc.) and parts of the piston cavity (piston, sleeve, piston rings) with two different grades of oils that best match the characteristics of the friction pairs. In addition, the separation of cavities protects the crankcase space of the engine from getting into it combustion products burst through the gas joint of the piston. This is especially important when the engine is running on high-sulphur heavy fuels, the combustion of which produces many chemically active components that adversely affect the service life of lubricating oil and can cause corrosion of engine parts and mechanisms;

- the separation of the sub-piston space from the crankcase cavity, in some cases, allows the sub-piston space of the working cylinder to be used to organise an additional operating process (double-acting engines), as a purge pump or for additional charge compression before the super-piston cavity is purged.

For larger *S*/*D* ratios, a crosshead engine will have a lower height than a trunk engine. This is explained by the fact that at large piston strokes the angle of deflection of the connecting rod from the vertical axis is limited by the diameter of the bushing, beyond which it must not extend. This makes it necessary to increase the length of the connecting rod. At the same time, in crosshead engines the length of the connecting rod, as a rule, slightly exceeds the diameter of the crank by the amount necessary to accommodate the crosshead assembly. Fig. 2.28 *a* shows the kinematic diagrams of an engine with *S*/*D* = 4 in trunk and crosshead designs. It can be clearly seen that in the crosshead version the engine, all other things being equal, will have a 32% lower height.

Fig. 2.27 *b* shows the relationships between connecting rod length and maximum centre distance (distance between the crank axis and the piston pin axis at the piston position at TDC) as a function of *S*/*D*. It is well seen that for *S*/*D* 0.8...1.6 the connecting rod length is proportional to the increase

For preliminary review

in the main ratio.

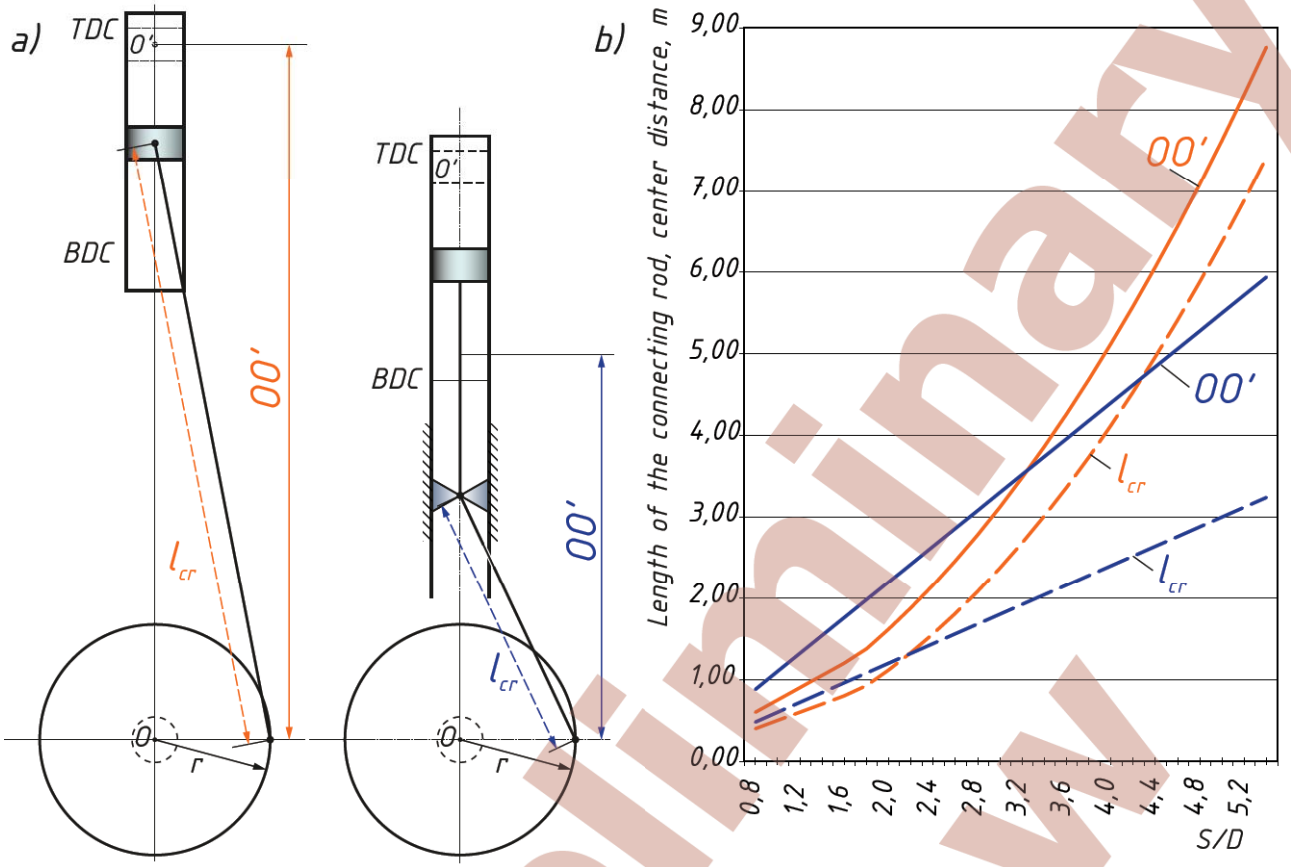


Figure 2.27 Kinematic diagrams of an engine with $S/D = 4$ in trunk (left) and crosshead (right) versions (*a*) and the dependence of connecting rod length and maximum centre distance (*b*) as a function of S/D for $D = 0.5$ m

In this case, the cylinder bore is sufficient to maximise connecting rod deflection from the vertical axis. The length of the connecting rod is determined by layout considerations, namely the need to provide sufficient centre-to-centre clearance between the crank axis and the piston pin axis at the piston's BDC position.

When the S/D increases above 1.6, the cylinder bore does not allow for the necessary connecting rod deflection, so the connecting rod length must be increased, which leads to an increase in the centre distance and height of the engine as a whole. The connecting rod length increases as a step function. In crosshead engines the connecting rod length does not depend on the cylinder diameter, therefore, in the whole range of S/D variation the linear dependence is maintained. From Fig. 2.27 *b* it can be clearly seen that for S/D greater than 3.2, the trunk engine becomes taller than the crosshead engine.

The disadvantages of crosshead engines include a more complex design of the gear train and a higher value of inertia forces due to the greater mass of moving parts.

This design scheme is suitable for both four-stroke and two-stroke operation. Earlier attempts were made to create four-stroke crosshead-type marine engines, for example, low-speed diesel engine K6V45/6G of MAN company. The experience of creation of such engines showed that they are characterised by poor mass-dimension indicators, and large masses of moving parts limit the possibility of their speed boosting. At present this scheme is used only for creation of two-stroke diesel engines of high power.

Crosshead engines usually have between 4 and 14 cylinders and are predominantly built as in-line engines. In this case they have the same advantages as in-line trunk engines. However, the needs of the fleet in modern container ships designed for simultaneous carriage of 10000...28000 containers (TEU) led to the fact that MAN considered the concept of building a V-shaped crosshead

For preliminary review

engine on the basis of the model K98MC-C with the number of cylinders from 12 to 18 and power from 55200 to 103000 kW. The general layout of such an engine is shown in Fig. 2.28. However, the construction of such engines was not achieved. It was decided that it would be more rational to install two in-line main engines on such ships.

Crosshead engines with oppositely moving pistons.

The scheme of such an engine is shown in Fig. 2.27 *e*, the general view of the crank mechanism in Fig. 2.29. Single-shaft engines with oppositely moving pistons have mainly shafts with three elbows for each cylinder. In this case, the two side elbows are at an angle close to 180° to the middle (main) elbow and are connected to the upper piston through a crosshead and long rods.

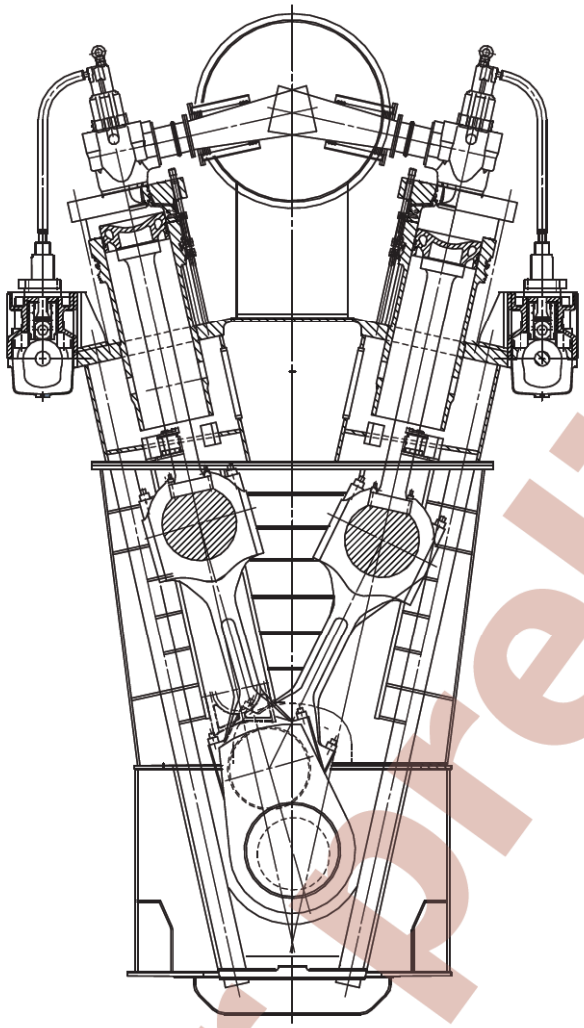


Figure 2.28 – Concept of the K98MC-C V-type crosshead engine (adapted from [8])

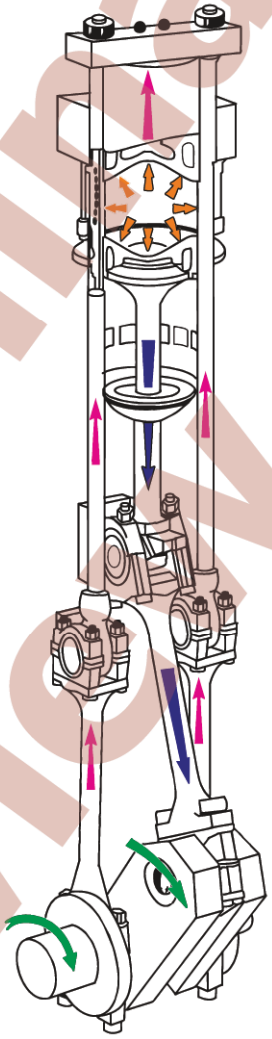


Figure 2.29 – General view of the crosshead engine with oppositely moving pistons (adapted from [9])

Engines of this type combine the advantages of crosshead engine and engine with oppositely moving pistons. The disadvantages of this scheme include the complexity of the crank mechanism design, large mass of moving parts. High heat intensity of the exhaust piston requires its intensive cooling.

Currently, the engines of this layout are not built, but in operation can still meet diesel engines of the English company Doxford.

It is interesting to note that in the history of marine engine building more complicated schematic solutions were used. For example, Harland & Wolf Engines in the 1930s produced double-acting crosshead engines with three pistons in the working cylinder, intake, exhaust and main pistons. The main piston was driven by the crank of the crankshaft, and the timing pistons by eccentrics located on the crank jaws. The double-acting principle allowed to increase the engine power by 1.5...1.8

For preliminary review

times with a relatively small increase in its weight.

At present, double-acting engines (usually two-stroke) are not produced, as they are characterised by complexity of construction, very difficult working conditions of the piston group, rod and other parts. In such engines it is difficult to ensure good quality of gas exchange processes and especially mixture formation in the cylinder cavity through which the rod passes. The use of supercharging makes it possible to obtain the required power in simpler engines of simple action.

Literature used when writing section 2:

1. Bilousov I., Bulgakov M., Savchuk V. Modern Marine Internal Combustion Engines. Springer International Publishing. 2020. – 385 p.
2. 645E3 Turbocharged Engine. Maintenance ManuAL. 7th Edition. Electro-Motive Division, General Motors Corporation, La Grange, Illinois, U.S.A. 1979 – 586 p.
3. Белоусов Е.В. Цикл Миллера и его реализация в судовых дизельных двигателях. / Белоусов Е.В., Чернявский В.В. // Двигатели внутреннего сгорания. – Харьков: изд. НТУ»ХПИ» – 2013. – № 1. – С 127-132.
4. Codana E., Vlaskosa I. Turbocharging medium speed diesel engines with extreme Miller timing. ABB Turbo Systems Ltd, Baden, Switzerland. 2004 – 20 p.
5. Белоусов Е.В. Новые подходы в организации рабочих процессов в судовых четырехтактных двигателях. / Белоусов Е.В., Белоусова Т.П. // Науковий вістник херсонської державної морської академії: Науковий журнал. – Херсон: вид. ХДМА, 2012.– № 2 (7). – С. 17-25
6. MAN L35/44DF. Project Guide – Marine Four-stroke dual fuel engine compliant with IMO Tier II. MAN Energy Solutions SE, Augsburg. 2018 – 4440 p.
7. MAN B&W S90ME-C10.5-GI-TII. Project Guide. Electronically Controlled Dual Fuel Two-stroke Engines. Edition 0.5, MAN Diesel & Turbo, Copenhagen, Denmark. 2014. – 393 p.
8. Container Ship Propulsion. MAN B&W Diesel. 1997. – 16 p.
9. Dragsted J. The first 50 years of turbocharged. 2-stroke, crosshead, marine diesel engines. CIMAC. Central Secretariat, Frankfurt am Main, Germany. 2013 – 94 p.

For preliminary review

CHAPTER 3

The design of marine internal combustion engines, the mechanisms and systems included in them. Forces and moments acting in marine engines

From a structural point of view, an internal combustion engine is a set of separate mechanisms and systems, each of which performs one or more functions. Given this, the division of the engine into separate mechanisms and systems is very conventional.

The crank mechanism includes the *cylinder-piston group* (*CPG*) (cylinder sleeve, piston assembly, cylinder cover) which absorbs the pressure of gases on the piston and the *motion group* which converts the reciprocating motion of the pistons into rotary motion of the crankshaft (basic parts, crankshaft, connecting rod, crosshead, piston rod, etc.).

The air-gas system supplies pre-cleaned air to the engine cylinders under atmospheric or increased pressure with pre-cooling of compressed air, and also removes combustion products from the engine, utilizes their heat and reduces exhaust noise. It includes (depending on the engine design) a *gas distribution mechanism*, a *system for purifying and supplying air to the engine (supercharging system)*, an *exhaust gas removal and exhaust noise reduction system, and an emission control system.*

The gas distribution mechanism is used to control the gas exchange processes of the slave cylinder in accordance with the set timing phases.

The engine air purification and supply system is used to take air from the ship's engine room or directly from the atmosphere, purify it from mechanical impurities and supply air to the engine cylinders. In most modern marine engines, this system includes a charge *air supercharging and cooling system* to increase the charge density at the engine inlet.

The *exhaust and exhaust noise reduction system* diverts combustion products from the engine into the environment.

The emission control system serves to reduce the emission of harmful substances in the engine exhaust gases.

It should be noted that engine mechanisms can combine several functions. For example, the pistons in the crank mechanism can partially or fully control the gas flows.

A fuel system that provides for the preparation and transportation of fuel and its subsequent injection into the engine combustion chamber under high pressure in a highly regulated manner.

The lubrication system is used to supply lubricating oil to all rubbing surfaces to reduce friction forces, cool rubbing parts and remove wear products from the friction zone.

Cooling system, serves to remove excess heat from the most heated in the process of engine operation parts, as well as from the working fluids.

Starting and reversing system, used to start the engine and change its direction of rotation.

Automatic regulation and control system, serves to maintain the specified mode of engine operation, to control the parameters of the operating process and the state of the main elements of the engine structure.

3.1 Fixed structural elements of marine engines

Regardless of design and purpose, the crank mechanism of any engine consists of fixed parts (backbone parts) and motion parts.

The fixed parts include: the foundation frame, the frame (crankcase), cylinder jackets, liners and their covers connected to each other. Together these parts make up the engine frame. Fixed elements of the crank mechanism of a four-stroke engine with a one-piece block-carter are shown in Fig. 3.1.

For preliminary review

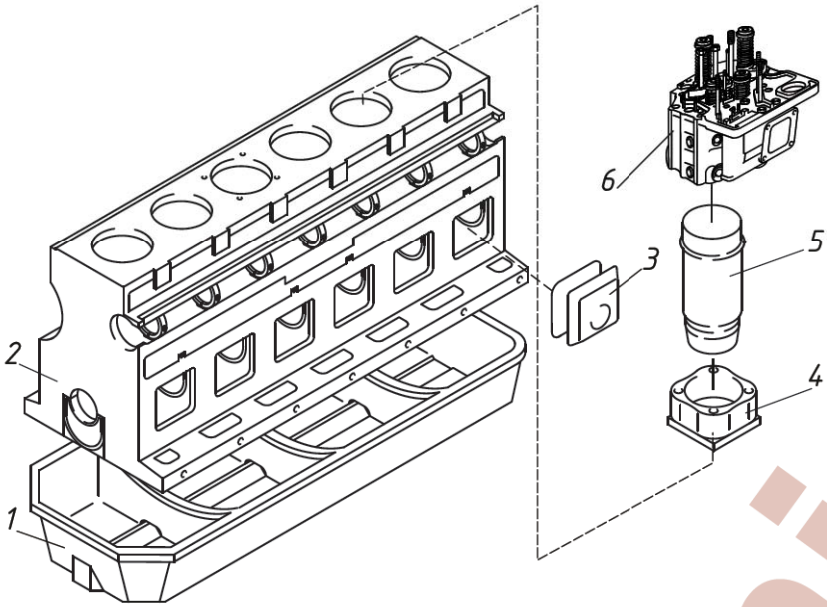


Figure 3.1 – Fixed elements of the crank mechanism of the four-stroke engine: 1 – crankcase pan; 2 – block crankcase; 3 – cooling jacket; 4 – cylinder sleeve; 5 – crankcase hatch cover; 6 – cylinder cover (adapted from [1])

3.1.1 Foundation frame

The foundation frame is the base of the engine. All other parts are fastened to it, and it itself is mounted and fastened to the ship's foundation with support flanges (Fig. 3.2 *a-d*).

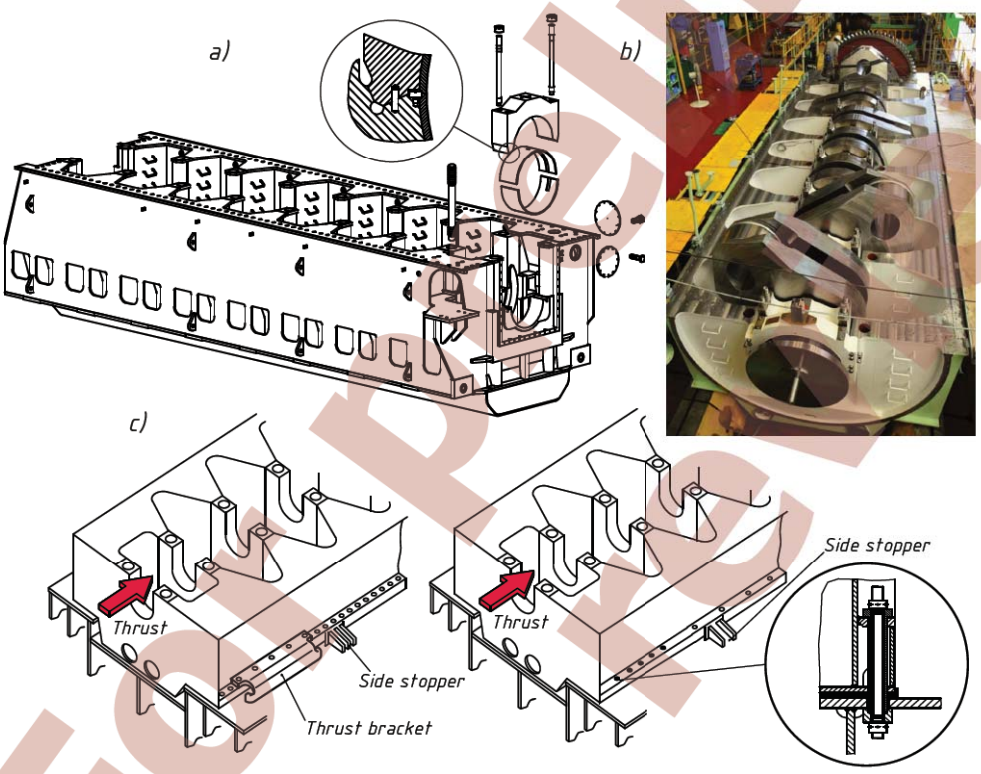


Figure 3.2 – Foundation frames and their fastening. *a* – foundation frame of MAN K90MC low-speed diesel engine; *b* – foundation frame with crankshaft of 7RTflex 82 engine; *c* – fastening of the engine to the ship's foundation by means of consoles; *d* – fastening of the engine to the ship's foundation by means of extended studs (adapted from [2, 3, 4])

For four-stroke as well as small two-stroke diesel engines, the foundation frames are predominantly cast. The most common material for these frames is cast iron, less often steel. For some high-speed engines, the base frame is made of aluminium alloys.

For powerful two-stroke diesel engines, the foundation frames are manufactured as all-welded or composite frames, mostly of the closed type. Compared to cast frames, welded frames are simpler and cheaper to manufacture and have a lower weight. In such frames only individual elements of complex shape are made by casting or stamping, and the rest are cut from sheet or shaped rolled steel. All structural elements are welded together and the frame is further processed as a single unit.

For preliminary review

Composite frames are usually made of two halves to simplify manufacturing and transportation. The individual frame parts are connected by means of prison bolts. The disadvantage of composite frames is reduced rigidity of the whole structure.

In a closed foundation frame, the spaces between the cross beams form the moth wells, between which oil flows through special holes in the cross beams. The system of longitudinal and cross beams forms the engine sump. In open frames, the sump is made separately from thin sheet steel and fastened with studs to the bottom of the frame. The sump is used to collect used lubricating oil dripping from the engine parts. The bottom of the sump is sloped for oil drainage into the outlet pipe.

The frame is supported on the ship's foundation by butt flanges with holes for fastening bolts.

In the vertical plane, the engine is centered using steel wedges or spherical shims. To center the engine relative to the shaft line, guide stoppers are installed along the machined side surfaces of the flanges (Fig. 3.2 *c*, *d*). In some cases, epoxy resin mastic is poured into the gaps between the mating surfaces of the base frame and the ship's foundation after the engine has been centered.

Special thrust brackets (Fig. 3.2 *c*) or bolted fasteners are used to secure the engine to the ship's foundation. In case of small engine sizes the fastening is carried out with short bolts. In case of larger sizes, extended bolts or studs with spacer tubes are used to increase their pliability during thermal expansion of the foundation frame (Fig. 3.2 *d*).

Part of the bolts in the aft part of the foundation frame are prisonally adjusted to the holes, the rest of the bolts are installed in the holes of increased diameter. This makes it possible for the foundation frame to expand when heated without disturbing its position relative to the shaft line.

Diesel generators and other auxiliary engines of relatively small sizes are attached to the ship foundation through special vibration absorbers (Fig. 3.3), which allow to significantly reduce the vibration transmitted to the ship's hull.

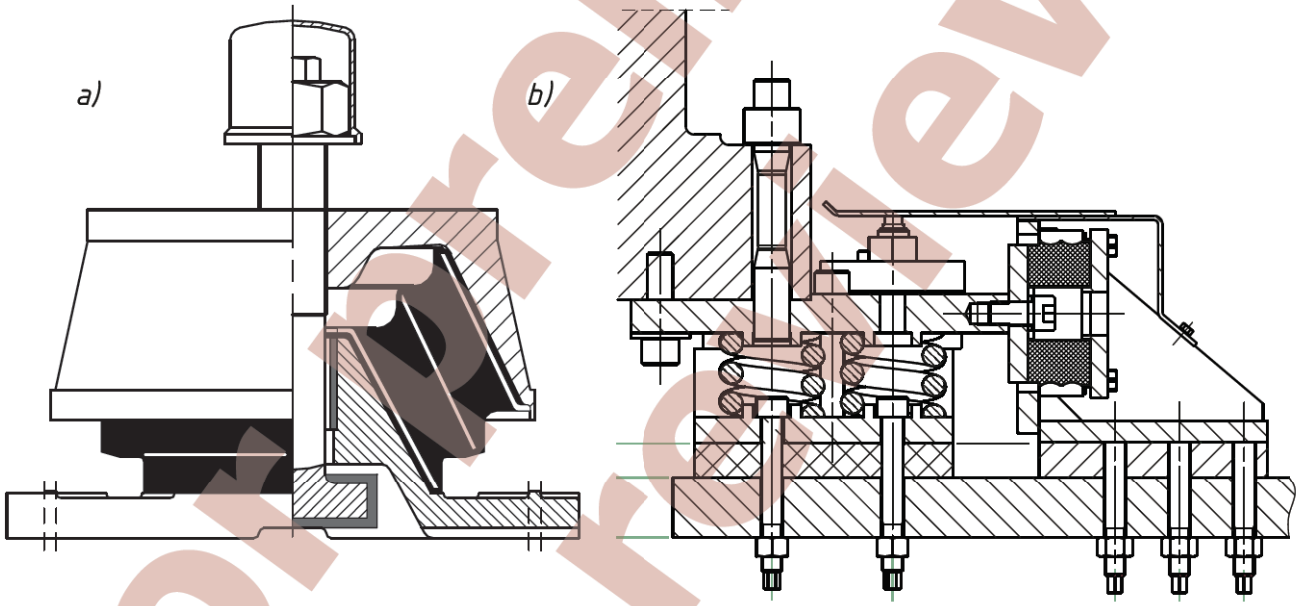


Figure 3.3 – MAN L16/24 engine shock absorber (*a*) and Wärtsilä L46 (*b*) [5, 6])

The main vibration absorber of a vibration damper is a rubber, spring or combined elastic element.

A bore called the crankshaft bed is made in the foundation frame, in which split frame bearings are installed to accommodate the crankshaft. The upper half of the bearing is secured by one or two covers that limit crankshaft movement (Fig. 3.2 *a*, *b*). The use of two paired covers simplifies removal and installation. The covers are attached to the foundation frame using threaded rods, screw or hydraulic screw jacks (Fig. 3.3 *a*, *b*)

Fig. 3.4 *a* shows the fastening of the top covers of the frame bearings by means of studs.

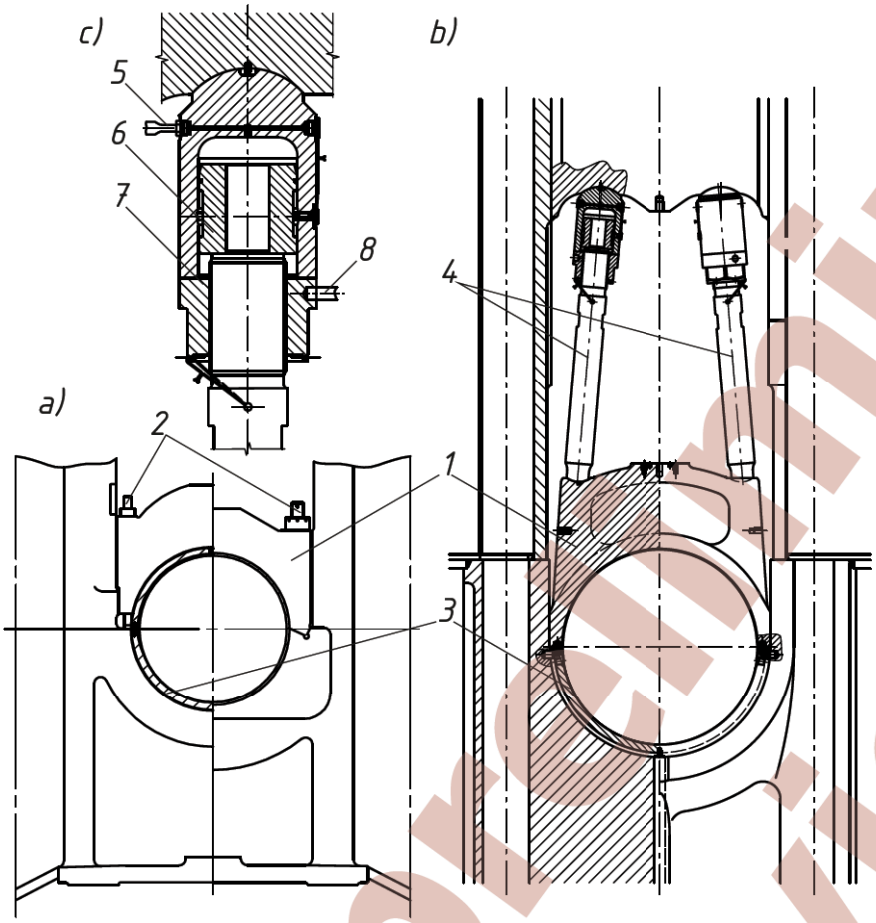
This design is quite simple, so it has found widespread use on all types of engines.

The disadvantage of this design is the need to have a large enough space above the cover to re-

For preliminary review

move it from the studs. Tightening of studs on most Low Speed Engines and Medium Speed Engines is done with hydraulic devices, which allows normalizing the tightening force.

Fastening of the frame bearing top cover with spacer hydraulic screw jacks is shown in Fig. 3.4 *b*. When mounting the cover, pressurized oil is supplied from the hydraulic power plant through connector *5* (Fig. 3.4 *c*) into the bodies of both jacks, which presses on the piston *6*. As a result, the threaded nut *7* is released, which is tightened with the help of a special rod *8*, after which the pressure in the jack is released and the position of the nut is fixed with a wire.



3.4 – Fastening of the upper cover of the frame bearing. *a* – by means of studs (MAN diesel engines of S, L and K series); *b* – by means of hydraulic screw jacks (Sulzer diesel engines of RTA series); *c* – device of hydraulic screw jack. 1 – bearing cover; 2 – studs; 3 – bearings; 4 – hydraulic screw jacks; 5 – oil inlet fitting; 6 – piston; 7 – compression nut; 8 – rod for cranking the nut (adapted from [7, 8])

3.1.2 Crankcase or crankcase box

The crankcase or crankcase box, depending on the engine design and its dimensions, is cast from steel or cast iron. For modern low-speed engines, the bed is made in the form of welded box structures made of sheet steel (Fig. 3.5). Boxed beds have high rigidity and a smaller number of bolted connections, which provides a good tightness of the crankcase. With a large length of the engine bed is made of two or three parts. In the lower part of the bed there is a buttress surface, which the bed rests on the foundation frame and is connected to it by bolted joints.

The bed is used to support the cylinder block. It connects the cylinder block to the foundation frame and forms an enclosed cavity for the crank mechanism, called the crankcase. In addition, the bed is the base member for the bulk of the engine auxiliaries, such as camshafts, fuel pumps, and valve train.

In order to prevent the destruction of the diesel engine on the side plates or hatch covers of the bed are installed spring-loaded safety valves 1 (Fig. 3.5), which are triggered automatically in case of explosions in the crankcase of the engine (Fig. 3.6).

Explosions may be caused by accumulated oil vapors, for the removal of which a crankcase ventilation system must be provided. The ventilation pipe, equipped with an oil separator and flame arrester, is usually routed from the crankcase cavity to the upper deck.

Vertical cast iron or steel plates reinforced on both sides with stiffening ribs, called parallels, are placed on the sides of the cross-braces of the bed to absorb the forces from the crosshead sliders.

For preliminary review

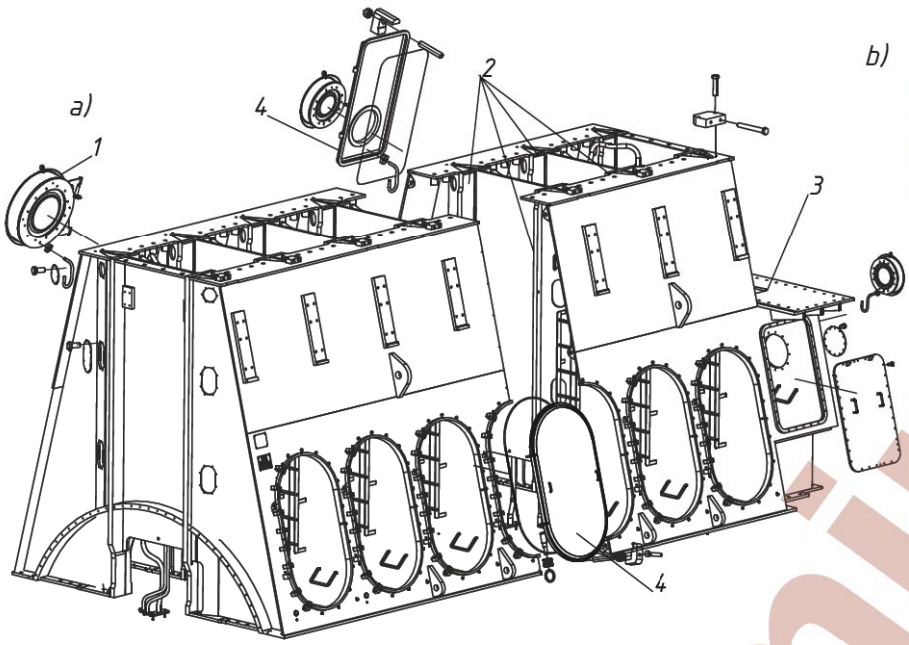


Figure 3.5 – Box-type bed of a modern low-speed diesel engine of MAN (*a*). 1 – crankcase safety valve; 2 – parallels; 3 – auxiliary shaft drive compartment; 4 – crankcase hatch cover. Wärtsilä RTA 58 T engine bed (*b*) (adapted from [9, 10])

For disassembly of bearings and inspection of motion parts, there are hatches on the side walls of the bed, which are closed with removable covers.

A compartment is provided in the end or middle part of the bed for installation of auxiliary shaft drive mechanisms and mechanisms.

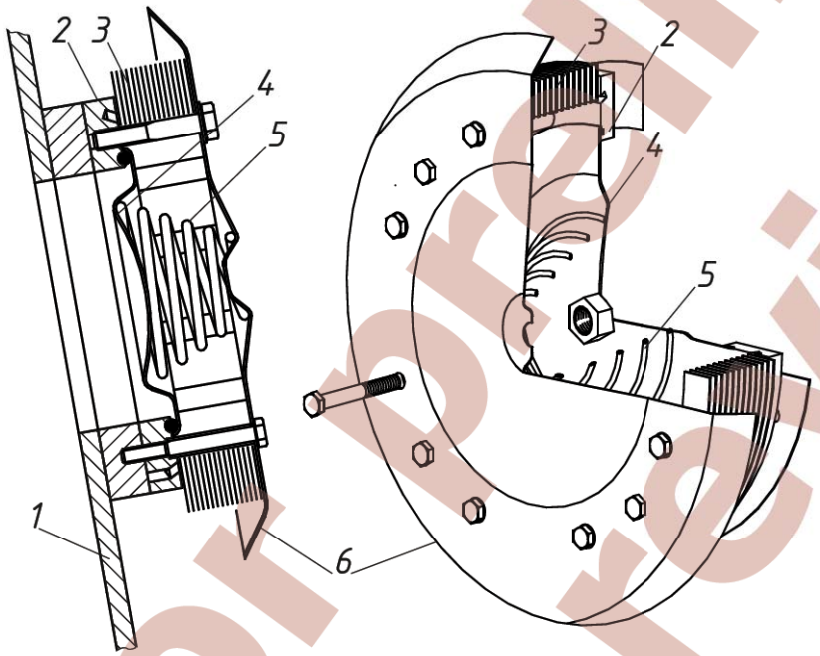


Figure 3.6 – Safety valve of the crankcase box: 1 – crankcase box; 2 – valve housing; 3 – flame arrestors; 4 – valve plate; 5 – pressure spring; 6 – protective casing (adapted from [9])

Modern four-stroke trunk -type engines widely use a block design of the frame, where the foundation frame as such is absent, and the bed is made as a single unit with the cylinder block (Fig. 3.7). The bottom of such a frame is covered with a light pallet for sealing the crankcase space and oil collection.

Modern methods of design with the use of three-dimensional modeling allow to obtain the most rational designs of block-carters having high rigidity with minimum weight of the structure.

Some engines have a supply of circulating oil in the sump, in which case the engines are called wet crankcase engines. To prevent oil foaming and to dampen oil surface vibrations, special grids or baffle plates are installed in the crankcase sump.

In such engines, the crankshaft bearing caps are attached from below (suspended type crank-

For preliminary review

shaft), receiving not only the force from the mass of the crankshaft and the parts of the motion group, but also the forces from the action of gases on the piston, as well as the inertia forces of the crank mechanism parts. For greater rigidity, the main bearing caps are fastened to the block crankcase by side bolts (Fig. 3.7 *b*).

The crankcases are usually cast in cast iron, steel or aluminium alloys, or welded from steel plates.

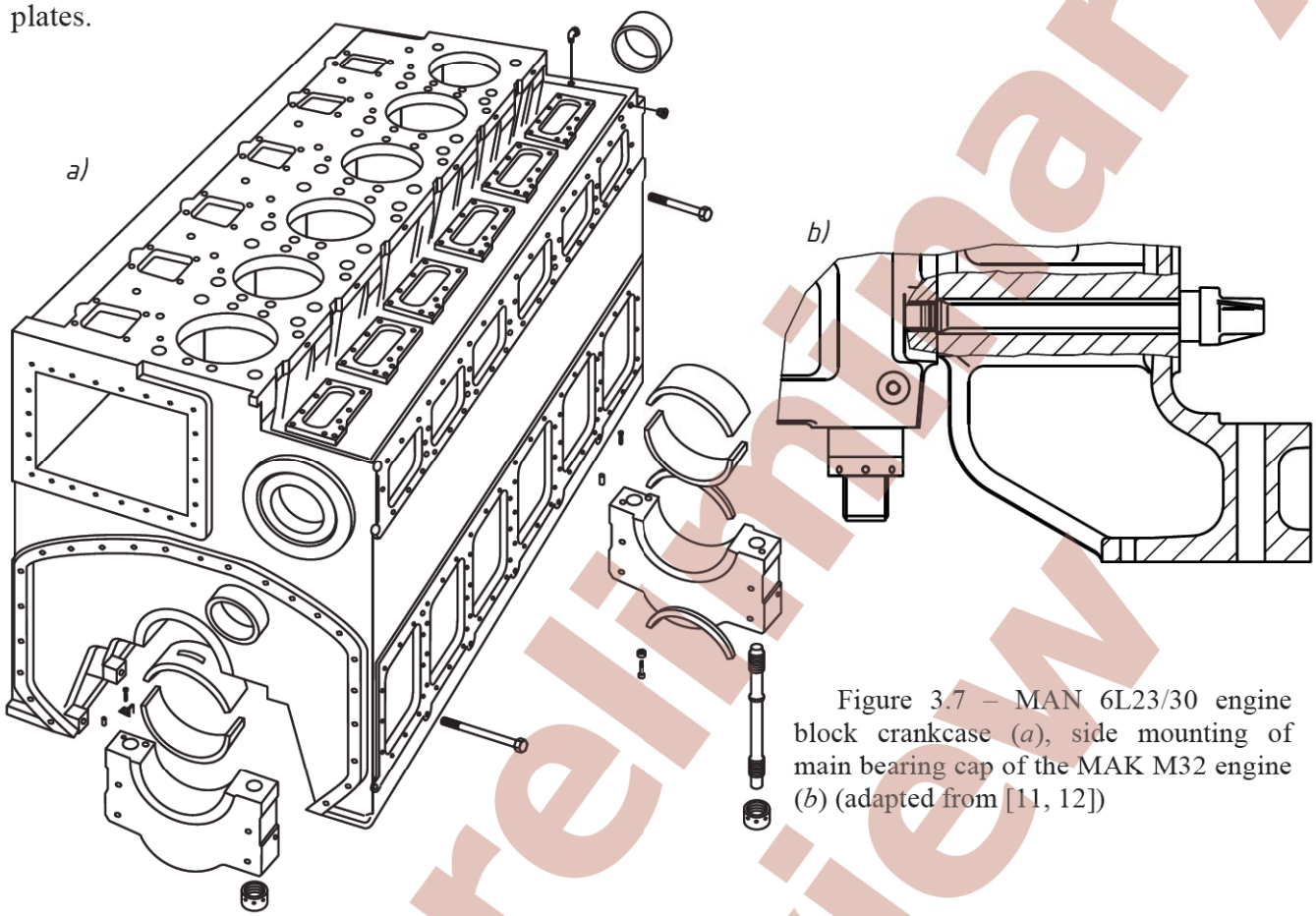


Figure 3.7 – MAN 6L23/30 engine block crankcase (*a*), side mounting of main bearing cap of the MAK M32 engine (*b*) (adapted from [11, 12])

3.1.3 Working cylinder

The working cylinder consists of an outer jacket 1 and cylinder sleeve 2 (Fig. 83 *a*). The sleeve is pressed into the bore of the cylinder jacket, between them there are cavities for cooling water circulation. In the jackets of two-stroke diesel engines, air cavities are provided for the charge air supply.

Jackets are manufactured as a monolithic block common to all or several engine cylinders. Block construction makes the engine more rigid, while reducing its length and weight. Jackets of powerful two-stroke engines are made separately for each cylinder and connected by bolts (Fig. 3.8 *b*). Sometimes jackets are mounted on the bed independently of each other. This scheme is especially common on medium- and high-speed engines designed to operate on heavy, high-sulfur fuels. To prevent acid corrosion in such engines cooled only the upper part of the hub, so the jacket of these engines are usually made in the form of a low spacer between the bed and the cylinder cover (Fig. 3.8 *c*).

Two-stroke engines have a special baffle, called a diaphragm, in the lower part of the engine crankcase cavity that separates the engine crankcase cavities from the sub-piston space. A special piston rod gland seal is installed in the diaphragm. On the intake side, the cylinder block has intake ports that connect the purge air cavities with the horizontally located charge air receiver. To enable cleaning of the back of the jacket space, the jacket is equipped with a neck with lids.

The sleeves, together with the piston and cylinder cover, form a closed volume in which the engine's operating process takes place. The sleeves are subjected to significant mechanical stresses from the pressure forces of the gases and thermal stresses from the temperature difference between the hot gases and the liquid in the bore space. In addition, the inner surface of the bushings are subjected to wear from abrasion by moving parts, and corrosive effects due to condensation on its walls

For preliminary review

of water vapor, which dissolve the oxidation products of sulfur and nitrogen to form mineral acids. The outer surface of bushings, bordering with water, is subjected to cavitation destruction due to micro-movements of the wall and oxidation by interaction with oxygen dissolved in water. Therefore, for reliable operation, the bushings must not only be sufficiently rigid, but also have good wear and corrosion resistance.

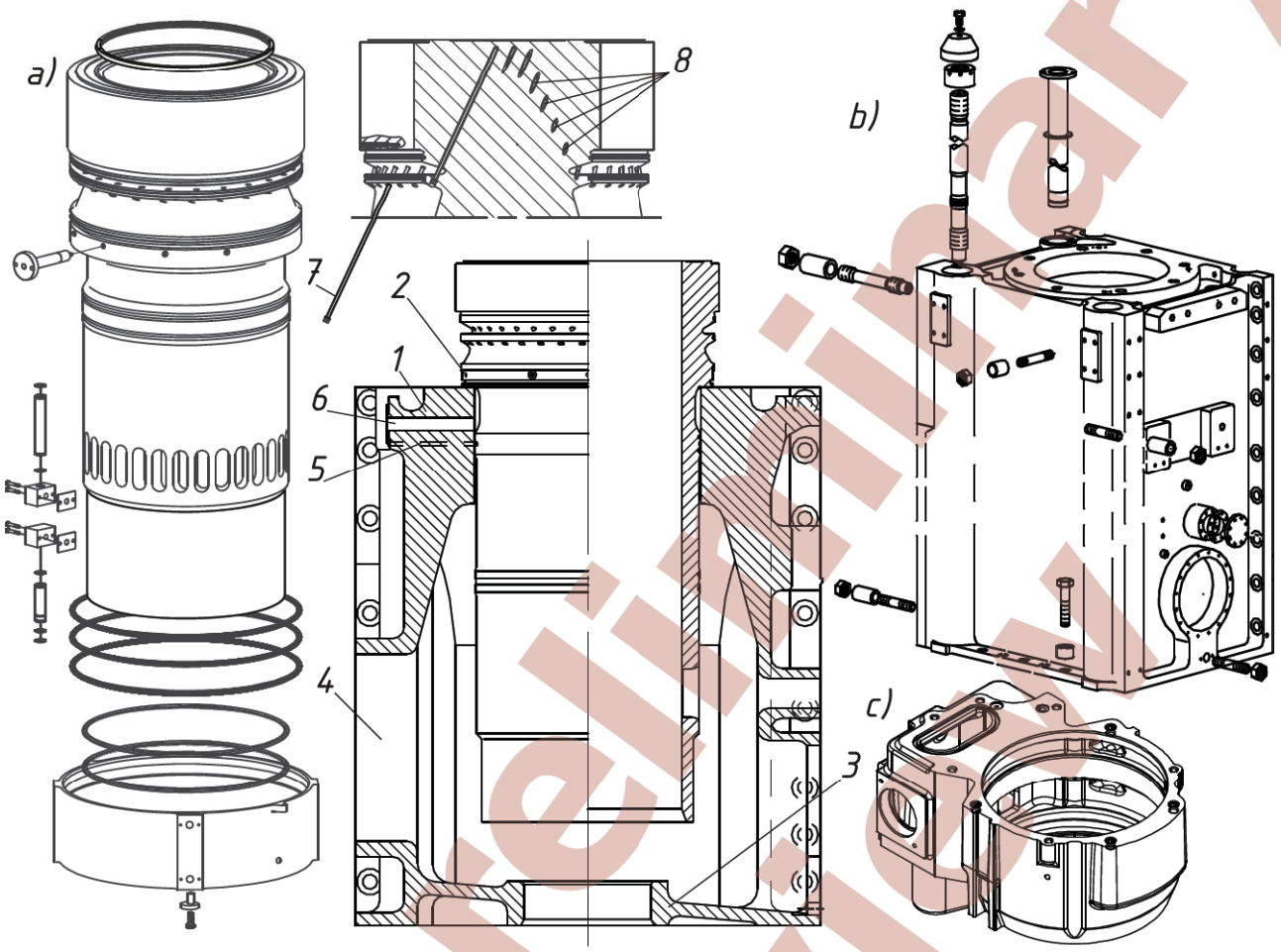


Figure 3.8 – Working cylinder of low-speed crosshead engine and, cooling jacket of MAN K90MC diesel engine (*a*, *b*), cooling jacket of high-speed diesel engine MAN L21/31 (*c*). 1 – cylinder jacket; 2 – cylinder sleeve; 3 – diaphragm; 4 – charge air inlet cavity; 5 – drain hole for seal condition monitoring; 6 – water supply for sleeve cooling; 7 – adjusting rod; 8 – fire surface cooling channels (adapted from [13, 14, 15])

The sleeves are usually made of pearlitic cast iron or cast iron alloyed with chromium and nickel. For medium and high-speed engines, cylinder sleeves are sometimes made of steel. The upper part of the sleeve is thickened, as this area is subject to the highest pressures. In most modern low- and some medium-speed engines, the upper part of the sleeve is made in the form of a massive flange inside which to relieve thermal stresses drilled slanted holes through which passes water from the cooling system (Fig. 3.9 *a*). The upper flange of the sleeve rests on the support shoulder of the jacket and is pressed against it by the cylinder cover (Fig. 3.8 *a*, 3.9 *a*).

On MAN engines of the L and S series, the cylinder sleeve rests on the cylinder block flange, with its upper part extending out of the cylinder block and encompassed by a thin jacket that creates a cooling cavity. To reduce the height of the jackets, in some designs the flange of the sleeve is supported by a spacer shoulder fitted to the butt face of the jacket or cylinder block (Fig. 3.9 *b*).

The bushing is inserted into the bore of the lower support belt with a gap, as it expands more than the cooling jacket when the engine is running. In addition, the sliding fit allows the bushing to expand axially during heating. This helps to reduce mechanical stresses and makes it possible to produce one-piece bushings of sufficiently long length.

For preliminary review

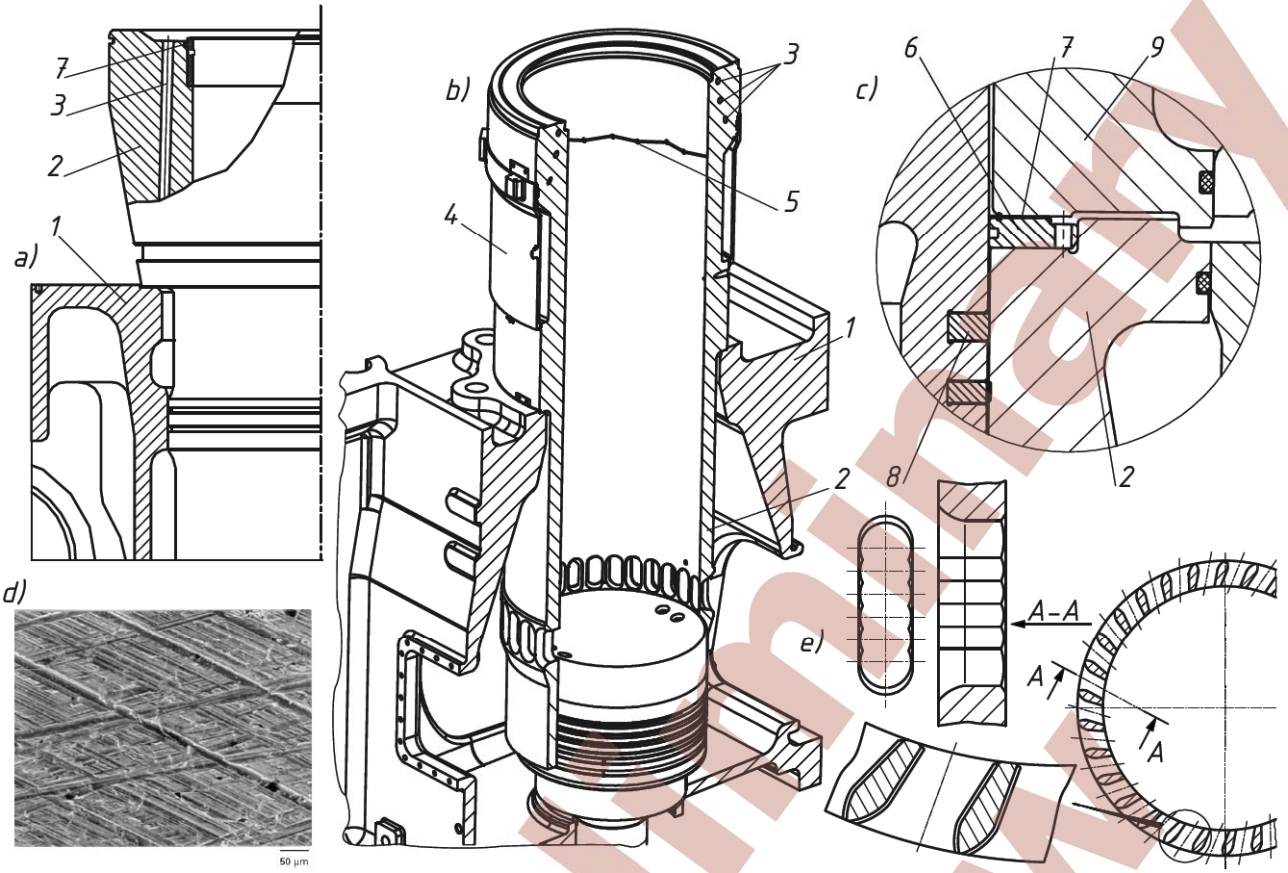


Figure 3.9 – Design and installation features of modern marine engines working cylinders: *a* – installation of cylinder sleeve of Wärtsilä L46 engine; *b* – installation of spacer between cylinder block and sleeve (Wärtsilä RT-flex 82 C series engines); *c* – installation of anti-polishing ring in MAN low-speed engines; *d* – microrelief on the working surface of cylinder liner; *e* – cross-section and direction of blow-through ports of Wärtsilä RTA series engines. 1 – cylinder block; 2 – cylinder sleeve; 3 – coolant channel; 4 – cooling jacket; 5 – oil distribution grooves; 6 – anti-polishing ring; 7 –steel gasket; 8 – piston rings, 9 – cylinder cover (adapted from [16, 17, 18, 19])

In some designs of high and medium speed engines, to reduce the height, part of the bushing is placed in the crankcase cavity, with slots made in the bushing to allow the connecting rod to pass through. Special slots are sometimes made in the upper part of the bushing to open the timing valves. When these design features are present, the bushing must be positioned in a specific position, which is achieved by means of locating pins.

A significant influence on the wear of the working surface of the sleeve is caused by carbon deposits formed on the cylindrical surface of the piston above the upper piston ring. These deposits contain mineral inclusions of high hardness and have an abrasive effect. To reduce wear of the sleeve surface on all modern engines, an anti-polishing ring is installed in the upper part of the sleeve (Fig. 3.9 *a*, *c*). It is a cylindrical insert in the upper part of the liner, which has an inner diameter smaller than the sleeve, but larger than the cylindrical part of the piston bottom. Thus, this ring forms a ledge at the level of the upper piston ring, which prevents the deposit of carbon deposits on the side walls of the piston head. In some engines, this insert also acts as a heat shield and is made of heat-resistant steel.

On two-stroke engines, the timing ports are cut into the lower part of the hub. In this place, to increase the cross-section of the bridges between the ports, a thickening is usually made, especially for engines with slotted purge schemes, where the exhaust port bridges are subjected to increased mechanical and thermal loads. In engines with straight-through purging schemes, the thickening of the belt in the area of the purge ports is not made, and the ports themselves are cut at an inclination to the axis of the cylinder sleeve to ensure the rotary movement of purge air in the cylinder (Fig. 3.9

For preliminary review

e). To seal the various cavities through which the cylinder sleeve passes, viton rubber rings are mounted on the cylinder sleeve and placed in specially located grooves. Usually the rings are installed in pairs to increase the reliability of the seal. In addition, the cavity between the rings has a drainage hole, leakage from which indicates a leakage failure of the upper sealing ring (Fig. 3.8 *a*). When sealing the exhaust gas cavity, copper belts rolled into grooves are used instead of rubber rings.

The inner surfaces of the bushings are hardened with high frequency currents, chromed, nitrided or hardened by other methods to increase wear resistance. To reduce friction between the surface of the bushing and the piston with piston rings, microrelief in the form of grooves at an angle to each other is applied on the inside of the bushing by honing (Fig. 3.9 *d*). These grooves retain oil and lubricate the piston rings.

The bushings of most high and medium speed engines are lubricated by oil aerosol contained in the crankcase and deposited on their surfaces. In some medium-speed diesels, oil is supplied to the cylinder liner through special oil distribution holes in the piston. In this case, pressurized oil is supplied to the piston through bore holes in the connecting rod. In low-speed engines, the crankcase cavity is separated from the sub-piston space, so in such engines oil is supplied to the bushings from a special dosing system called lubrikatornoy. In this case, the bushing has special drilled holes for the installation of oil inlet fittings. In order to better distribute the lubricant, there is a wavy groove in the area of the outlet holes of the fittings.

The outer surface of the cylinder sleeve may have special ribs to increase rigidity and heat exchange area, especially for engines with oppositely moving pistons. To protect against corrosion, the outer surface of the sleeves can be coated with special paints or varnish.

3.1.4 Anchor links

Anchor links are long studs (bolts) passing through the foundation frame, bed and cylinder jackets, linking all elements of the frame into a single rigid structure. The anchor links are tightened with forces exceeding the maximum combustion pressure. This ensures that the frame components are relieved of the tearing forces caused by gas pressure on the piston and cylinder cover, so that the engine frame components work only in compression. This is important both for cast iron structural elements, which have 3...5 times less strength in tension than in compression, and for welded steel structures, where the compression stress protects welded seams from fracture. Anchor ties are made of high quality carbon or alloy steels and ground along the entire length to relieve stresses in the material.

The anchor ties are divided into long and short ties. Long anchor ties are usually through-type, with the upper nuts located on the cylinder block a and the lower nuts located on the lower plane of the cross beams of the frame bearing bed (Fig. 3.10 *a*, 3.11).

This design has the greatest rigidity, but it has a significant disadvantage – all elements of the frame structure are constantly subjected to high compression forces. The increase in operating pressures in the cylinders of modern engines has led to the fact that the required tightening forces cause deformation of the engine frame and, above all, the crankshaft bed. This requires an increase in the rigidity of the main elements of the frame and, as a consequence, an increase in their weight. For this reason, MAN prefers short anchor links for the compact engine versions (MC-C), where the lower end of the stud is screwed into a threaded hole in the trough of the frame bearing bed flange. For more uniform force distribution, the number of anchor links is doubled (Fig. 3.10 *b*, 3.11 *d*).

To increase the resonant frequency of the studs, they are secured against axial vibrations by means of jacking bolts or intermediate bands (Fig. 3.11 *a*, *c*).

In some engine designs, to reduce the length of the anchor rods and for ease of installation, composite anchor ties are used by connecting the rods with threaded couplings.

The correct tightening of the anchor links determines the stability of the frame bearings and the position of the cylinder sleeves in the diesel engine block. For this reason, the studs must be tightened evenly and in the sequence specified by the manufacturer.

For preliminary review

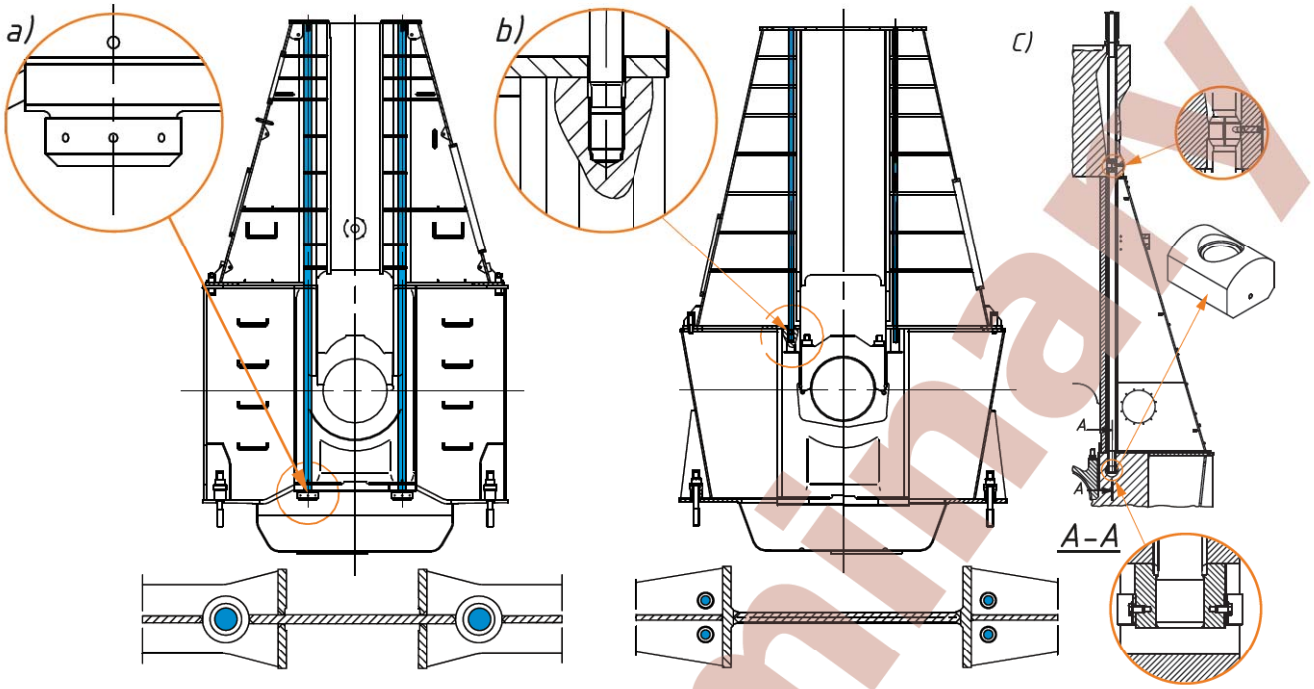


Figure 3.10 – Types of anchor ties used in marine internal combustion engines: *a* – long anchor ties; *b*, *c* – short paired anchor ties (adapted from [20])

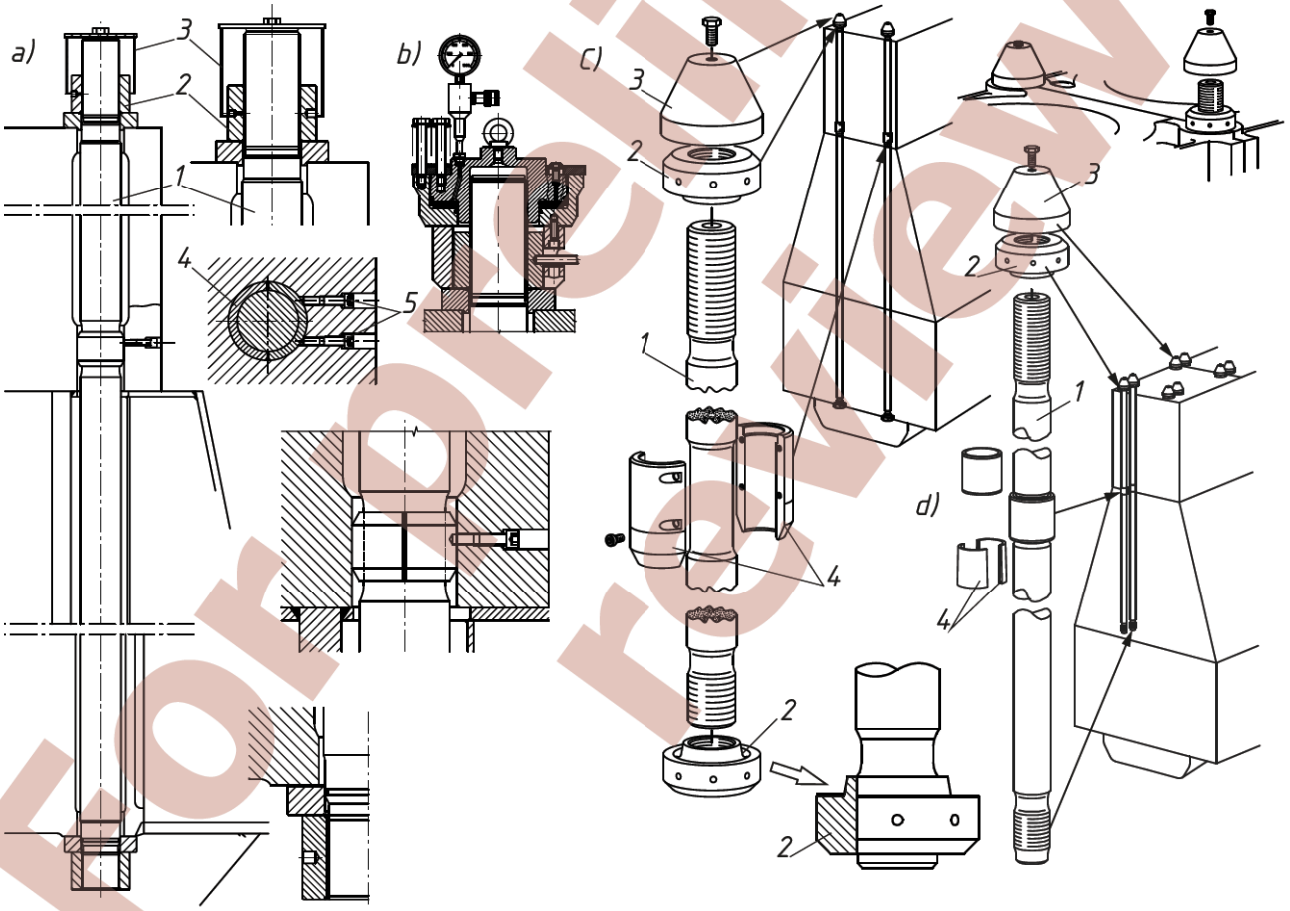


Figure 3.11 – Design features of Wärtsilä's RTA series Low Speed Engine anchor ties (*a*) and MAN K, L, S MC (*c*) and MC-C (*d*) series; *b* – hydraulic jack for tightening anchor ties: 1 – stud; 2 – nut; 3 – protective cap; 4 – bandage; 5 – fixing screws (adapted from [2, 18, 21])

On modern marine diesel engines the anchor links are tightened with special hydraulic jacks (Fig. 3.11 *b*). For this purpose, the hydraulic jack piston is screwed onto the thread above the nut. After that, the hydraulic station injects oil with pressure of 40...50 MPa under the piston. As a re-

For preliminary review

sult, the stud is lengthened, and between the nut and washer there is a gap, by the amount of which the nut is turned. After the studs are tightened and the jacks are removed, to prevent damage or clogging of the threads, the protruding part of the stud is covered with a special cap.

3.1.5 Cylinder covers

The cylinder covers together with the piston bottom and the sleeve surface form the cylinder working cavity and, when the piston is at TDC, the combustion chamber of the engine (Fig. 3.12 *a*).

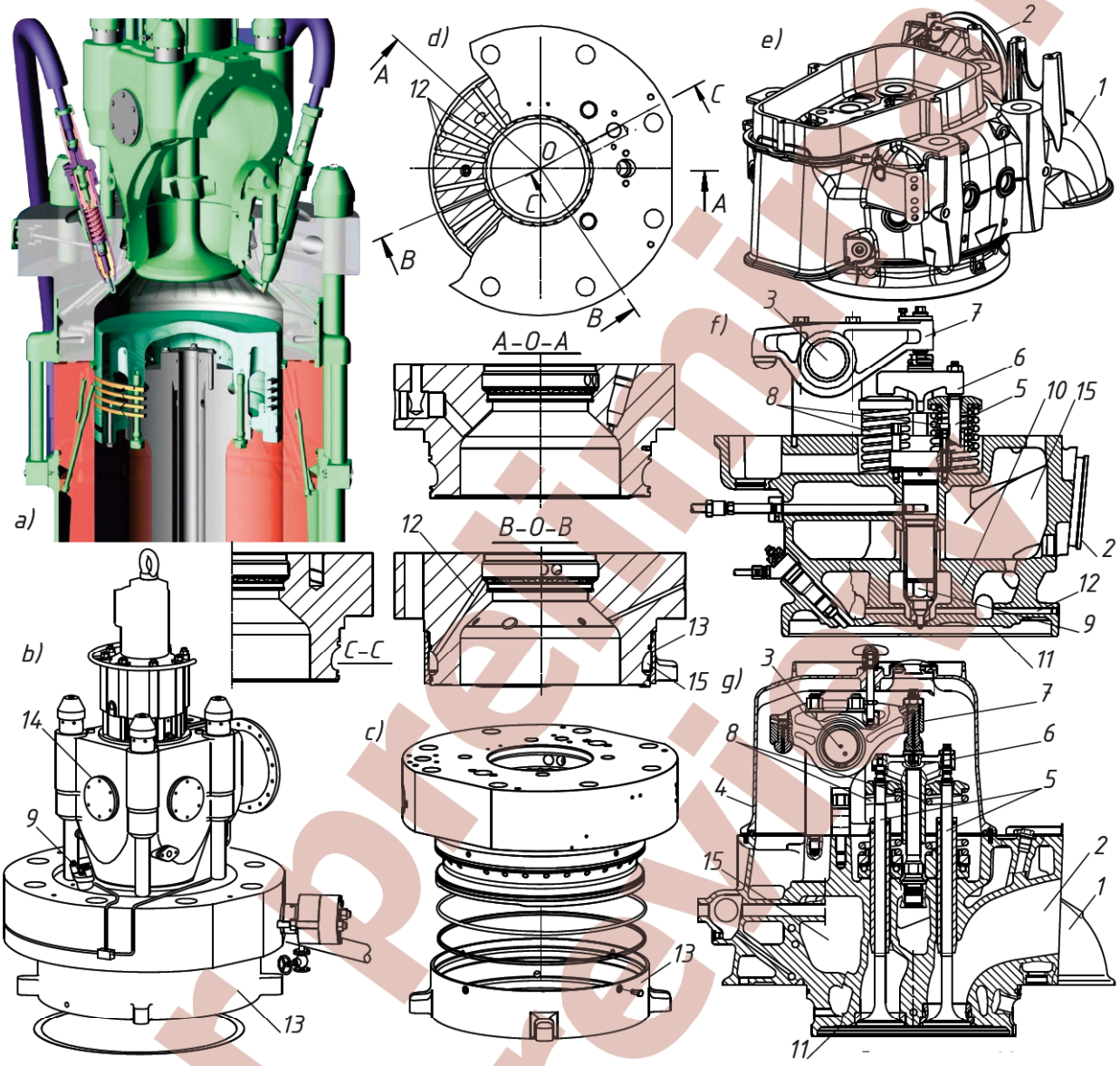


Figure 3.12 – Features of the design of cylinder covers of marine engines: *a* – general layout of the combustion chamber of the K98MC two-stroke engine from MAN; *b* and *c* – general view of the cylinder cover (engines of the K, L, S series from MAN); *d* – top view and cross-section of the cylinder cover of a two-stroke engine with a direct-flow valve purge circuit from MAN; *e* – general view of the cylinder cover of the MaK 32C four-stroke engine; *f* – cross-section of the cylinder cover of a 46 series diesel engine from Wärtsilä in the plane of the injection valve; *g* – cross-section of the cylinder cover of the four-stroke MaK 32C engine in the plane of the valves: 1 – inlet pipe; 2 – outlet pipe; 3 – axis of the valve drive lever; 4 – valve box cover; 5 – gas distribution valve; 6 – valve drive traverses; 7 – valve drive rocker arm; 8 – valve springs, 9 – nozzle; 10 – power bottom; 11 – fire bottom; 12 – drilling for cooling the fire bottom; 13 – water collector ring; 14 – exhaust valve housing; 15 – water collector (adapted from [16, 21, 22, 23])

They take the force from the pressure of the gases and are subjected to high temperatures. Therefore, the cylinder covers material must have both high mechanical strength and be heat-resistant. Cast iron and alloyed steel are used for manufacturing of covers of medium speed engine and high

For preliminary review

speed engine. In case of low-power high-speed engines, aluminium alloys are sometimes used for manufacturing of covers. Structurally, cylinder covers are made both individually for each cylinder and for a group of cylinders, usually assembled in one block. In this case, the cover is called a cylinder block cover. This design is characteristic mainly for small engines. Cylinder caps of modern small-speed engines are made mainly of steel forgings with subsequent machining. The general view of the cylinder cover of a low-speed engine with valve-slotted blow-by is shown in Fig. 3.12 *b*, *c*.

Cylinder covers of two-stroke engines with straight-through valve purge and especially covers of four-stroke engines are quite complex products. Since with relatively small dimensions in them must be placed inlet and exhaust valves, channels for air inlet and exhaust gas, injectors, starting and safety valve, as well as indicator valve (Fig. 3.12 *e*, 3.13). In addition, the surface of the cover facing the working cylinder cavity, called the fire bottom, must be intensively cooled, for which purpose water circulation cavities are provided inside the cover. In addition, such a saturated structure must have a sufficiently high mechanical strength, e.g. in the low-speed MAN K98MC engine the pressure force of the gases on the cover can reach values equivalent to 1,100 tons. The temperature of the gases on the side of the fire bottom can exceed 1800°C, which explains the need for its intensive cooling.

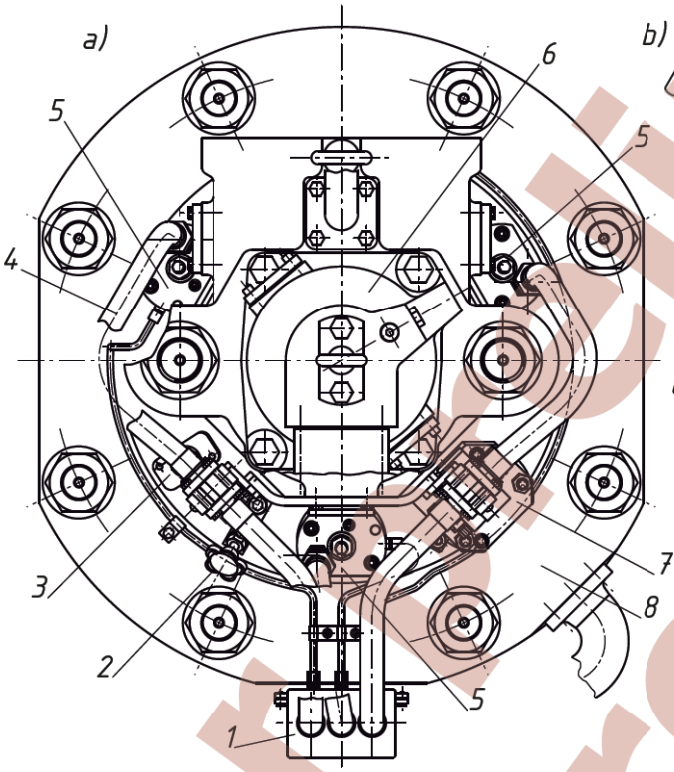


Figure 3.13 – Equipment of the cylinder cover of the two-stroke diesel engine RTA 72UB (*a*); indicator tap (*b*); relief valve (*c*). 1 – fuel distributor for injectors; 2 – indicator tap; 3 – relief valve; 4 – high pressure pipe; 5 – injection valve; 6 – exhaust valve drive; 7 – air start valve; 8 – cover flange (adapted from [18])

The cover is cooled with water coming from the space of the cylinder cooling jacket through special transition pipes or drillings in the cylinder liner flange, and is discharged from the highest point of the cover to avoid the formation of steam and gas locks. To more intensively remove heat from the fire bottom, two-level cooling is sometimes used in the covers of four-stroke engines. In this case, a second power bottom is placed above the relatively thin fire bottom, which is connected to the fire bottom through racks formed by wells of gas distribution valves, injectors, etc. (Fig. 3.12 *f*). The main fluid flow moves along the rear side of the firing bottom. In the case of high-force medium speed engine and high speed engine to increase rigidity, the fire bottom is made thick enough, and temperature stresses are relieved by passing water through a system of drilled holes in the bottom, as close as possible to the firing surface (Fig. 2.13 *f*). In order to prevent overheating of the exhaust valve, medium and high-speed engines operating on heavy fuels usually have a separate circuit for supplying coolant to the grooves in the valve seats.

To clean the cooling cavities, the cover is provided with hatches closed with covers or plugs. In

For preliminary review

the covers of modern medium speed engines, the cavities for cooling the fire bottom are made in the form of radial drills through which water is supplied for cooling not only the housing, but also directly to the exhaust valve seat (Fig. 2.13 *d*). All drills open into a water manifold formed by a groove in the cylinder cover and a ring placed on this groove (Fig. 3.12 *d*). The low temperature of the cap and the intense cooling of the lower part of the nozzle cup made it possible to use fuel instead of water for its cooling, as was the case in earlier designs. Undoubtedly, such feature makes these engines more convenient in operation.

Steel or red-copper rings or composite gaskets are used to seal the joint between the cylinder cover and the sleeve, which are crimped when tightened to seal the joint. In some engines, sealing is achieved by precise fitting of the mating planes.

The medium speed and high speed engine covers are attached to the unit jacket by means of studs, the number of which usually ranges from 4 to 8. The number of studs is based on the design of the cover. Increasing the number of studs leads to a more even distribution of loads, but makes the construction more complicated. For two-stroke engines, the studs pass through special holes in the cover flange. For engines with a straight-valve blow-by scheme, the number of studs usually does not exceed 8. For engines with other purge schemes, which do not have an exhaust valve, the number of studs can be up to 20, which allows for more uniform compression of the gasket.

Moving parts. The moving parts of the crosshead diesel engine include the piston assembly, rod, crosshead, connecting rod and crankshaft with flywheel; the trunk engine has no rod and crosshead.

3.2 Movable structural elements of marine engines

Moving parts. The moving parts of a crosshead diesel engine include the piston assembly, rod, crosshead, connecting rod and crankshaft with flywheel; A trunk engine does not have a rod or crosshead.

3.2.1 Piston

The piston is the most responsible element of any engine design, it takes the force of gas pressure and transmits it through the motion parts to the crankshaft, provides sealing of the gas joint with the cylinder guide sleeve, and in two-stroke engines it also controls gas flows. The piston operates under conditions of high, mechanical, dynamic and temperature loads, which imposes a number of special requirements to its design, material and manufacturing technology.

The metals used for piston manufacture must have high mechanical strength and wear resistance at sufficiently high temperatures, and for efficient heat dissipation from the firing surface they must also have good thermal conductivity. By design, pistons can be one-piece or composite. In composite pistons, the individual parts are usually made of different materials.

Regardless of the intended use, all pistons consist of a head and a guide part called the trunk or skirt. In turn, the head consists of the bottom, which limits the working space of the cylinder, and the side surface, on which the grooves for compression and oil rings, called caps, are located. While the purpose of piston heads in all engines is the same, the trunk, depending on the type of engine, can perform different functions. In engines with direct transmission of forces from the piston to the connecting rod, the trunk part is especially developed, because it fixes the position of the piston relative to the cylinder axis. In addition, trunk perceives and transmits to the cylinder wall normal forces arising from the deviation of the connecting rod from the cylinder axis. It is in view of this design feature that the engines of this layout are called trunk engines. In two-stroke engines, the trunk part serves, among other things, to control the gas flow, while in crosshead engines, in which normal forces do not act on the piston, the gas distribution function is the main one.

Fig. 3.14 shows the design of pistons of modern two-stroke low-speed engines, which are made of composite: the head is made of heat-resistant molybdenum or chrome-molybdenum steels, and the trunk is made of alloyed cast iron or aluminium alloys. The composite design of the piston allows you to reduce the consumption of expensive heat-resistant steels, reduce its weight, and, if necessary, allows you to replace the piston head.

For preliminary review

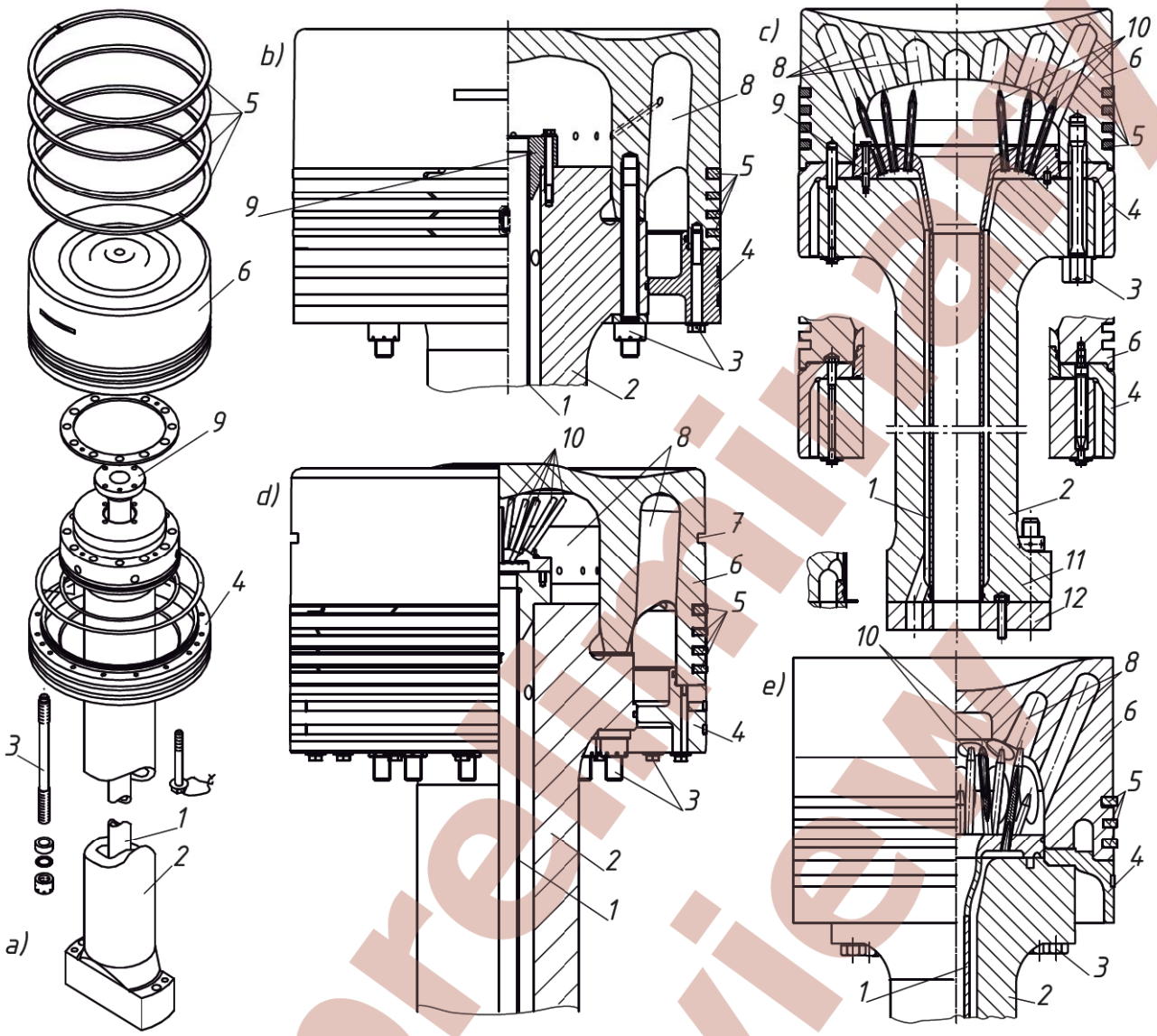


Figure 3.14 – Pistons of marine low-speed two-stroke engines: *a* – basic elements of piston and its rod of engines of series K, L, S of MAN; *b* – piston with rod of engines of series K, L, S of MAN; *c* – piston with rod of engine RTA 58 of Wärtsilä; *d* – piston of engine of series K, L, S Mark 9 of MAN; *e* – piston of engine UEC 50 LSE of Mitsubishi. 1 – drain pipe; 2 – piston rod; 3 – fasteners; 4 – piston trunk; 5 – grooves with piston rings; 6 – piston head; 7 – thermal barrier; 8 – cooling cavity, 9 – cooling pipe flange; 10 – cooling nozzles; 11 – rod mounting flange; 12 – regulating spacer (adapted from [24, 25, 26, 27])

During engine operation, the piston bottom is subjected to high mechanical and thermal loads simultaneously, resulting in rather contradictory requirements for its design. On the one hand, mechanical strength increases with increasing bottom thickness, on the other hand, this leads to large temperature differences, resulting in high internal stresses that can destroy the head material. In addition, the piston head closes the combustion chamber, and its outer surface should have a shape that provides the most optimal mixing. However, given that the thermal effect on the piston of a two-stroke engine occurs twice as often as that of a four-stroke engine, the heat stress of the latter is much higher. To reduce the heat transfer area, the firing surface of the piston should be as small as possible. In practice, these requirements are met by selecting an optimal bottom design and its intensive cooling. To increase mechanical strength in the bottom design, additional ring supports (Fig. 3.14 *b*) reducing the diameter of the unsupported part of the head, stiffening ribs (Fig. 3.14 *d*) or thickened bottom with a system of inclined drills (Fig. 3.14 *c*, *e*) are used.

Water or oil is used to cool the piston heads, which is supplied to the internal piston cavity. Wa-

For preliminary review

ter cooling is more effective, as water has higher heat capacity (4.187 kJ/(kg×K)) than oil (2.094 kJ/(kg×K)), but it is difficult enough to create a reliable system of seals allowing water supply to pistons. Therefore, recently all low speed engine manufacturers are gradually switching to piston cooling with oil, which, unlike water, is not so dangerous to leakage.

Coolant enters the piston through a channel in the rod (Fig. 3. 14 *a*, *b*, *c*, *e*), or through special telescopic rods.

Coolant is supplied to the bottom of the piston by pumping along the back of the head, by agitation during the piston stroke from TDC to BDC (Fig. 3.14 *b*, *d*), or by jet cooling, through a system of nozzles, each of which delivers oil to the corresponding head bore (Fig. 3.14 *c*, *e*).

Cooling of pistons is organized so as to maximally reduce the temperature in the area of piston ring grooves, especially the upper one. For this purpose, the head sometimes has a special cooling cavity located above the grooves, and to divert heat flows from the area of the installation of rings, in the upper part of the head make a ring groove 7, acting as a thermal barrier (Fig. 3.14 *b*). Given the differences in heating temperatures of the piston head and cylinder sleeve, which leads to different thermal expansion, the diameter of the head is selected so that between its cylindrical part and the sleeve was a gap sufficient to compensate for this difference without jamming the piston.

The piston head is attached to the rod flange by bolts or studs, which are made long enough to increase pliability. Sometimes spacers are used to lengthen the studs. In addition, the flange of the coolant inlet and outlet pipe and the piston trunk are attached to the rod flange. In MAN low speed engines of K, L, S series, the piston trunk is attached directly to the piston head (Fig. 3.14 *b*). On the trunk s of some pistons are installed special anti-seize rings made of lead or tin bronze. The rings are rolled into special grooves and protrude above the trunk surface by 0.1...0.4 mm.

Since all modern low speed engines have straight-valve blowdown, the trunk of such engines has only *a* guiding function (Fig. 3.14 *a*, *b*, *c*, *e*), so it is made relatively short. This makes it possible to significantly increase the piston stroke without increasing the engine height. For example, in the engine G80ME-C9.5 this figure is 2.73 meters. The increase in long stroke has made it possible to optimize the shape of the combustion chamber, namely, to increase its height and move the main volume into the cylinder cover cavity. As a result, it was possible to reduce the heat stress of the piston by reducing the area of the firing surface, and the main heat pressure was transferred to the firing bottom of the cylinder cover, which is much easier to effectively cool. To protect the cylinder sleeves from thermal effects in modern low speed engines, the volume of the combustion chamber is formed only by the surfaces of the piston and cylinder cover. For this purpose, the piston head at the position in the TDC significantly protrudes into the cap cavity (Fig. 3.15). Thus, at the moment of combustion and during part of the piston stroke, as long as the temperature of gases remains high enough, they are in contact only with the surfaces of the cap and piston.

As a result of the technical solutions discussed above, it was possible to achieve a significant increase in the specific performance of engines and their reliability in operation.

In trunk type engines (Fig. 3.16) for the pivot connection of the piston with the connecting rod on the trunk provided two troughs called bosses, which are made bore for the installation of a piston pin or sliding bearing under it.

In one-piece pistons, the bosses are designed as an extension of the head, which provides additional structural rigidity (Fig. 3.16 *b*).

The uneven distribution of temperatures and masses determine the complex geometry of the trunk piston. To compensate for the relatively large linear expansion of the piston head associated with higher heating temperature, the forming surface of the head is made smaller in diameter than that of the trunk, giving the piston a step, cone or barrel shape (Fig. 3.17 *a*, *b*, *c*, *d*).

To compensate for linear expansion of more massive bosses, the piston base is made in the form of an ellipse with a major axis perpendicular to the piston pin axis (Fig. 3.17 *e*). Differences between diameters D', D'' and semi-axes of the ellipse are chosen such that the piston acquires the most correct cylindrical shape at the steady thermal mode.

In composite pistons of trunk engines (Fig. 3.16 *a*, *c*, *d*, *f*), the head is usually made of alloy steels or cast irons, and the trunk part is made of cast iron or aluminium alloys. To increase the ri-

For preliminary review

gidity of the trunk, its lower part is thickened. The head is connected with the bottom by means of extended studs or bolts. At Caterpillar high speed engines some spread are composite pistons, in which the trunk part is connected with the bottom through the piston pin, which is simultaneously inserted into the holes on the trunk and on the bosses, made as a continuation of the head (Fig. 3.18).

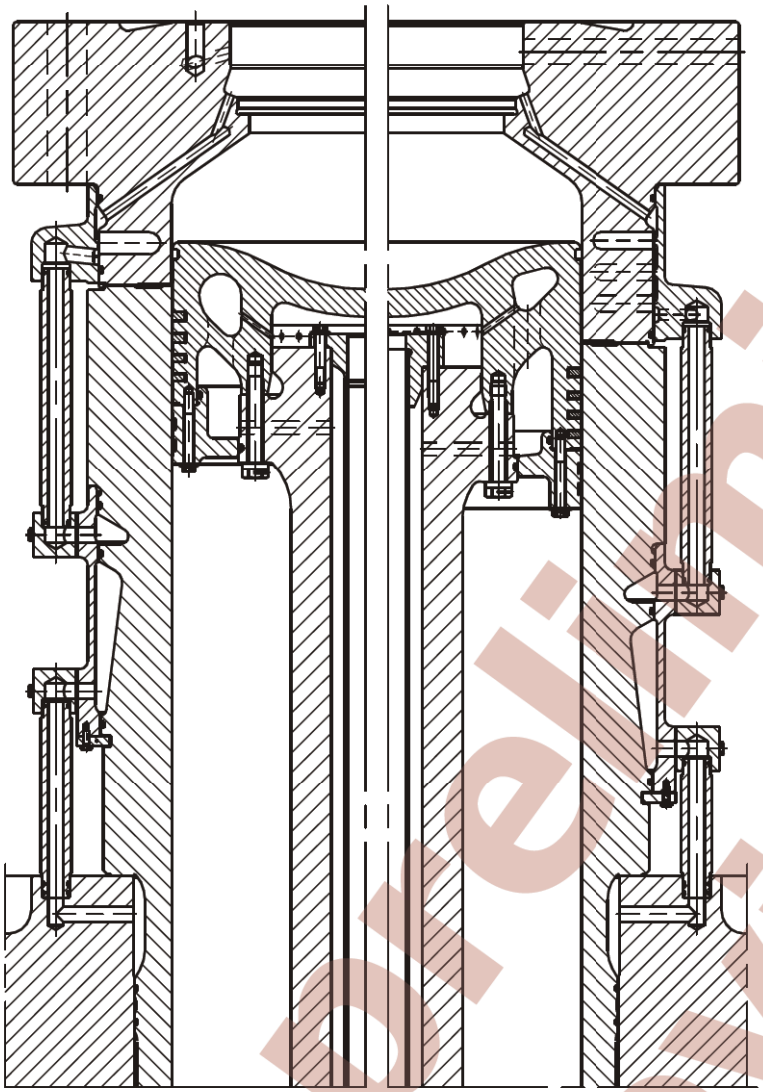


Figure 3.15 – Options for the combustion chamber of MC and MC-C engines with a low (left) and high (right) piston head (adapted from [28])

To strengthen the piston bottom, stiffening ribs are placed on it (Fig. 3.16 *c*), and the unsupported part is given a vaulted shape with high bearing capacity. Under the fire bottom of the piston in some designs is a second bottom, which closes the cooling cavity and acts as a thermal shield protecting the upper connecting rod bearing from excessive heating (Fig. 3.16 *a*, *b*, *f*).

Cooling of piston heads of modern high-force trunk engines is carried out by oil supplied from the general lubrication system. For this purpose, cavities are made in the head, which have the form of annular cavities, as well as radial or inclined drills 7 (Fig. 3.16 *b*, *c*, *d*, *f*). For intensive cooling of piston rings, these cavities are located above the caps.

Oil is supplied to the piston cooling cavity through a spring-loaded fitting 10 from the upper head of the connecting rod or through a drill hole in the piston pin, where it enters from the engine lubrication system through a drill hole in the connecting rod stem (Fig. 3. 16 *f*). Oil can be supplied in a continuous flow, providing flow cooling, or in individual portions. In the second case, the cooling cavity is not completely filled, and cooling occurs by agitating the oil when the piston changes direction.

From the cooling cavity, oil is drained into the lateral recesses on the outer side of the piston near the bosses, which are called coolers. In addition, a part of oil through the system of calibrated channels 4 is fed directly to the cylinder liner, directly under the oil rings.

For preliminary review

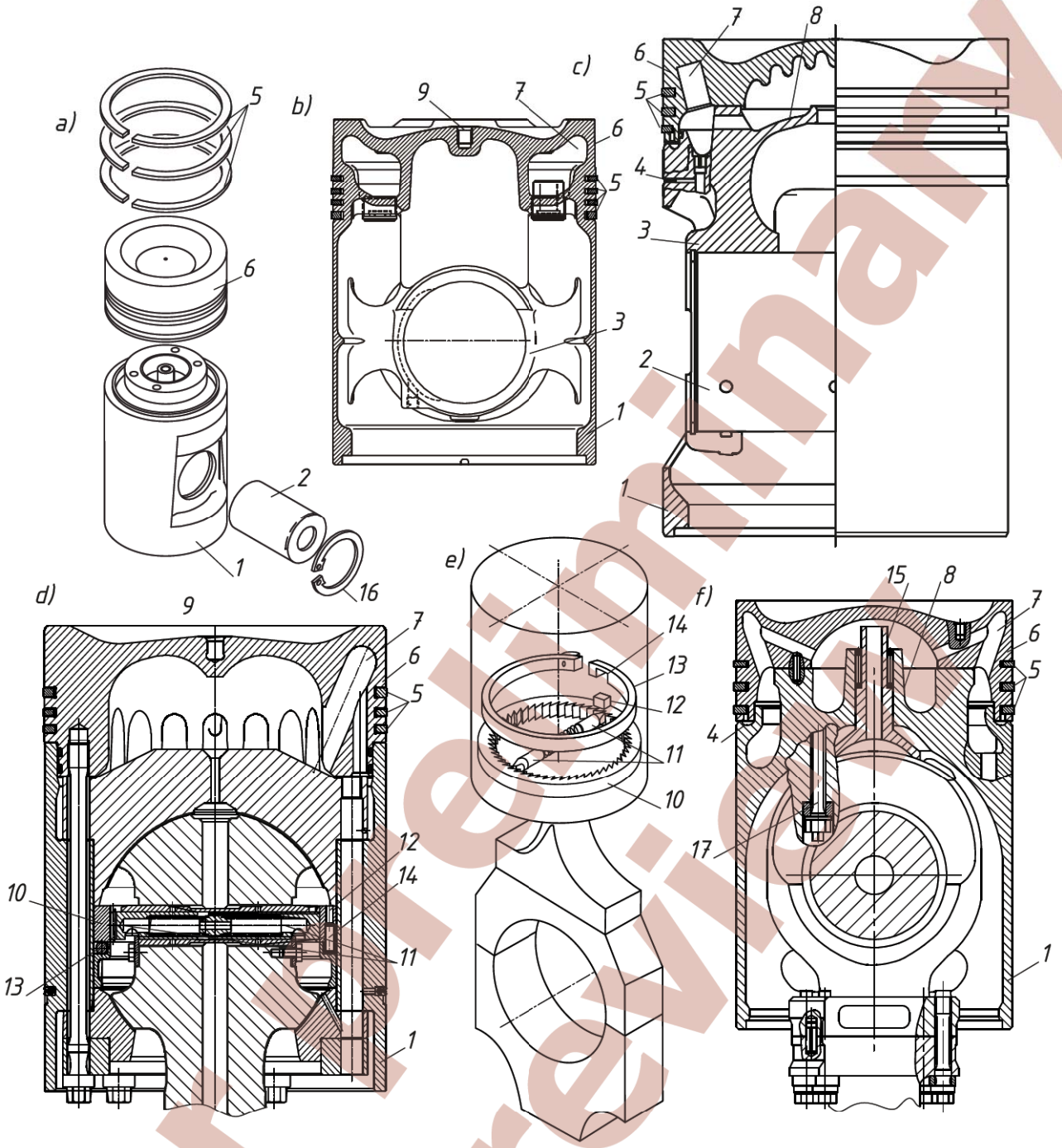


Figure 3.16 – Pistons of marine medium- and high-speed engines: *a* – general view of a piston of four-stroke engine L16/24 of MAN; *b* – piston of engine L23/30 of MAN; *c* – piston of engine 46 series of Wärtsilä; *d* – piston of engine Z40 of Sulzer; *e* – scheme of mechanism for piston turning of engine Z40; *f* – piston of four-stroke diesel engine M32 of MAK. 1 – trunk; 2 – piston pin; 3 – piston bosses; 4 – oil supply channel to the cylinder sleeve; 5 – caps with piston rings; 6 – piston head; 7 – cooling cavity; 8 – heat shield, 9 – threaded hole for eye-bolt; 10 – ratchet wheel; 11 – pawls; 12 – protrusion of the ratchet wheel; 13 – ring spring; 14 – slot in the spring; 15 – cooling oil supply connection; 16 – snap ring (adapted from [15, 16, 29, 30, 31, 33])

In engines with moderate forcing of the operating process jet cooling of pistons is used, when the bottom of the piston is washed with a jet of oil from a special nozzle installed in the upper head of the connecting rod. Uncooled pistons are used in high speed engines, which are usually made of aluminium alloys with thermal conductivity twice as high as that of cast iron and steel. This allows to simplify the design of the piston and reduce its weight. The disadvantage of using aluminium alloys for manufacturing pistons is their relatively low mechanical strength and a large coefficient of

For preliminary review

linear expansion. The latter circumstance forces to make increased thermal clearances between the piston and the sleeve.

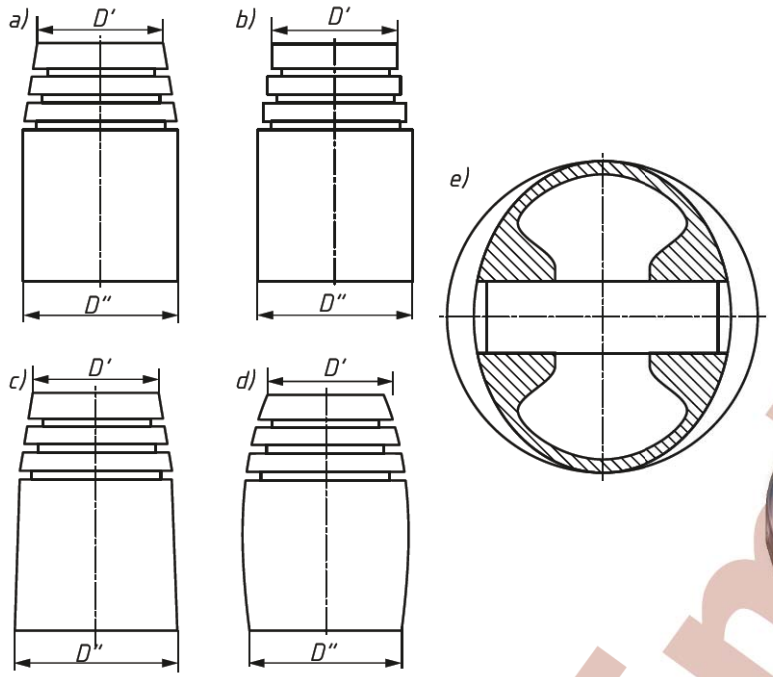


Figure 3.17 – Piston outline of tronc engine: *a* – cone head; *b* – stepped head; *c* – conical head and piston trunk; *d* – the side surface of the piston has a curvilinear profile with maximum diameter near the piston pin axis; *e* – piston cross-section along the piston pin axis

Figure 3.18 – Composite piston of C9 series engine of Caterpillar company

In two-stroke trunk engines, where the piston trunk acts as a timing spool, the piston trunk must have a solid cylinder shape to ensure that the entire area of the purge ports is covered to prevent lubricating oil from entering the purge receiver. A general view and cross section of such a piston of the EMD 710 engine of Electro-Motive Diesel is shown in Fig. 3.19.

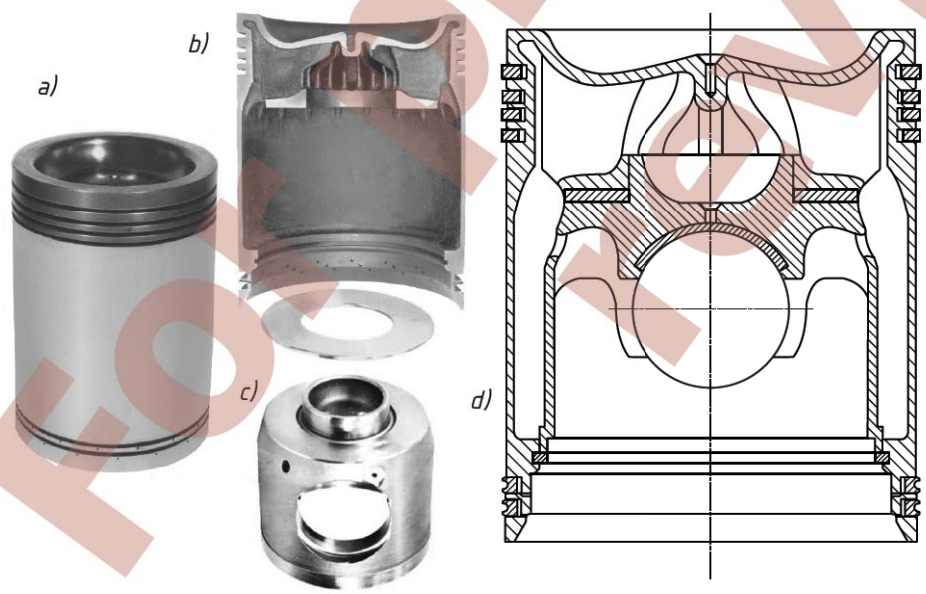


Figure 3.19 – General view (*a*, *b*, *c*) and cross section (*d*) of the piston of the Electro-Motive Diesel EMD 710 two-stroke tronc engine (adapted from [34])

3.2.2 Piston pin

Piston pins (Fig. 3.20-3.22) are made of low-carbon or alloy steels by subjecting their surface to nitriding, cementing and surface hardening. To improve bearing operating conditions and to relieve internal stresses in the hardened layer, the pin surface is ground.

For preliminary review

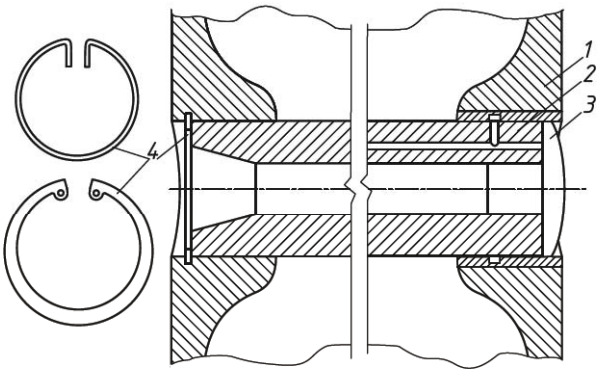


Figure 3.20 – Designs of piston pin, methods of its installation and axial fixation. 1 – bosses; 2 – piston pin; 3 – plugs; 4 – spring retainers

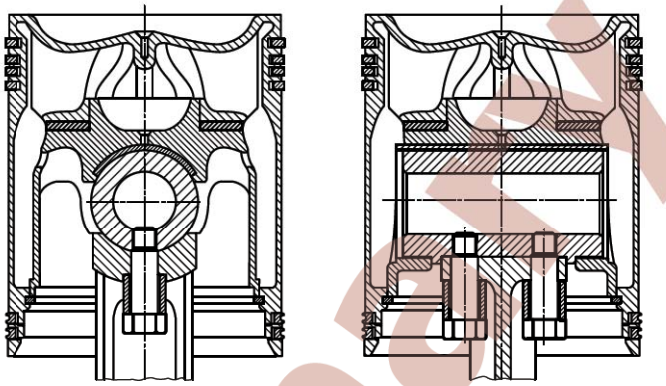


Figure 3.21 – Mounting the pin to the connecting rod stem with Electro-Motive Diesel EMD 710 extended bolts (adapted from [33])

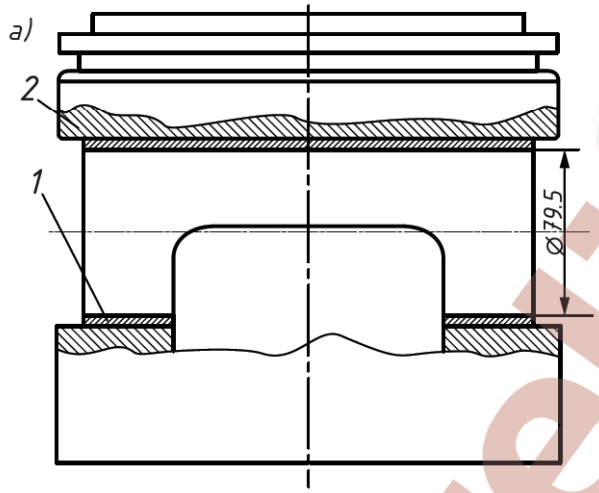


Figure 3.22 – Design of piston pin bushing (*a*) of General Electric's 7FDS two-stroke high-speed trunk engine and its general view (*b*)

Piston pins are designed in the form of a cylinder. In order to reduce the mass inside the pin, a bore with a cylindrical or conical shape is made. The conical bore (Fig. 3.20 *a*) makes it possible to distribute the stiffness of the pin in each cross-section in proportion to the acting loads, and thus to reduce its mass as much as possible, which is especially important for high-speed engines. In order to supply pressurized oil for bearing lubrication or piston cooling, a system of radial and axial borings is made in the pins of some engines (Fig. 3.20 *b*).

An interesting design solution was used by Sulzer in its Z series engines, which were originally designed and produced as two-stroke engines with a solid cylindrical forming surface of the tronc. The pistons of the engines in this series were fitted with a piston turning mechanism. Later the engines were redesigned for four-stroke cycle, but the piston and connecting rod pivot design was retained.

To provide the necessary degrees of freedom, the piston is connected to the connecting rod by means of a ball bearing (Fig. 3.16 *d*, *e*), which has a drill hole perpendicular to the cylinder axis and slightly displaced relative to it. Two spring-loaded pawls are located in the drill hole, which rest on the ratchet wheel freely rotating in the groove of the ball bearing (Fig. 3.23).

Due to the eccentricity of the pawls, when the connecting rod swings, the ratchet wheel makes a rotational movement, which is transmitted to the spring through a tab entering the recess on the ratchet wheel. As a result, the spring stores energy, which it gradually releases by rotating the piston.

The absence of bosses promotes a more uniform distribution of piston material over the piston diameter. For this reason, pistons of this type have a circle rather than an ellipse at the base. In addition, the rotation of the piston allows for a more even distribution of temperatures across the piston surface, thus improving operating conditions. The above factors reduce the clearances between the piston and the cylinder sleeve. Reduction of clearances reduces oil consumption and reduces shock

For preliminary review

loads during piston shifting, increasing the life of the cylinder. The use of a ball bearing allows the piston to be more accurately centered in the sleeve and the specific pressures to be more evenly distributed across the sleeve surface.

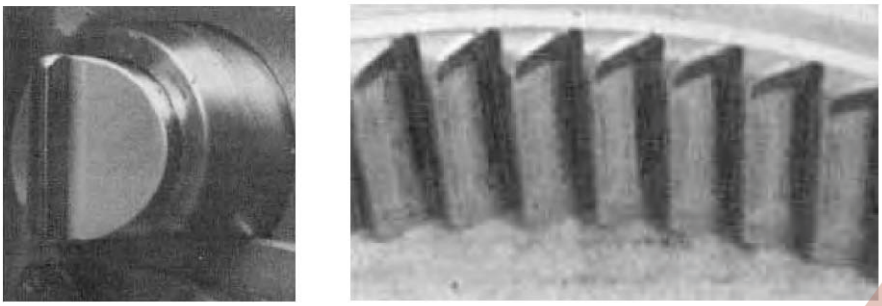


Figure 3.23 – Pawl (left) and fragment of the ratchet wheel of the piston rotation mechanism (adapted from [33])

3.2.3 Piston rings

For normal operation, the piston cavity must be sufficiently sealed and, at the same time, the piston must be movable relative to the cylinder sleeve. To fulfill these rather contradictory requirements, pistons are fitted with sealing elements called compression rings (Fig. 3.24).

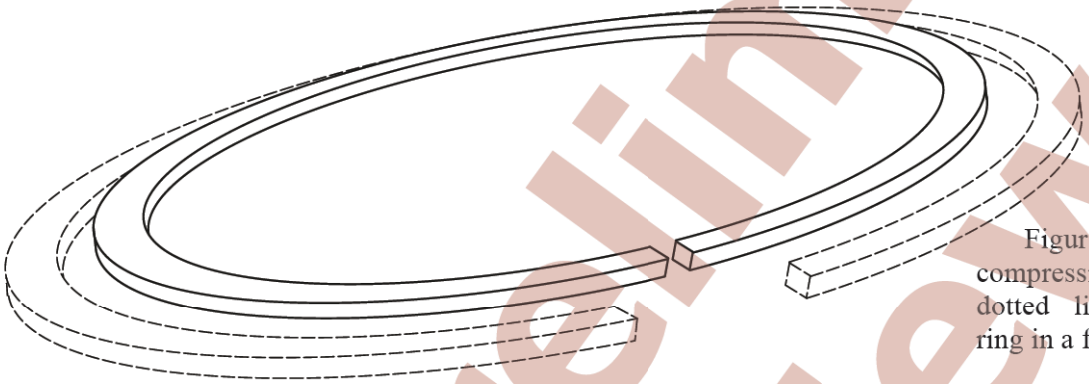


Figure 3.24 – Piston compression ring. The dotted line shows the ring in a free state

The number of compression rings depends on the engine design and usually ranges from 2 to 6. On modern engines, the number of rings and their height are tended to be reduced to reduce friction forces in the CPG. Structurally, the ring is a cut elastic element, which in radial section can have one of the shapes shown In Fig. 3.25 (*a-f*).

In the free state all piston rings are ellipse with a cut-out segment called a lock. When installed in the sleeve, the ring shrinks to a round shape and fits tightly to the surface of the sleeve (Fig. 3.24). The gap in the lock after its installation in the bushing should be sufficient to compensate for the thermal expansion of the ring during heating. Therefore, the value of the setting gap in the lock depends on the cylinder diameter and the maximum possible temperature of the ring.

Lock gaps are made straight (Fig. 3.26 *c*) or profiled (Fig. 3.26 *a*, *b*). On modern low speed engines the lock of the upper ring is made in the form of a complex profile shown in Fig. 3.26 *a*. Such locks provide a good seal of the gas joint, but are difficult to manufacture. All subsequent rings on the low speed engine are made with an oblique cut (Fig. 3.26 *b*), directing the slope of the joint in different directions. On four-stroke engine rings, the locking slot is made straight (Fig. 3.26 *c*). This lock shape provides less sealing efficiency, but is characterized by simplicity and reliability. The end gap between the ring and the cap must be sufficient to prevent the ring from jamming when the piston deforms under mechanical and thermal loads.

The rings are pressed against the working surface of the cylinder sleeve by their own elasticity. In this case, the maximum pressure is in the lock area. In order to reduce wear, various forms of pressure distribution patterns around the circumference are used. Particularly on low-speed engines, rings are used with the ends slightly bent inward. This correction serves to prevent the rings from hitting and breaking on the edges of the purge and exhaust ports.

In addition to the elastic forces, a significant role in pressing the rings, especially the upper rings, is played by the pressure of the gases, which acts on the back side of the ring.

For preliminary review

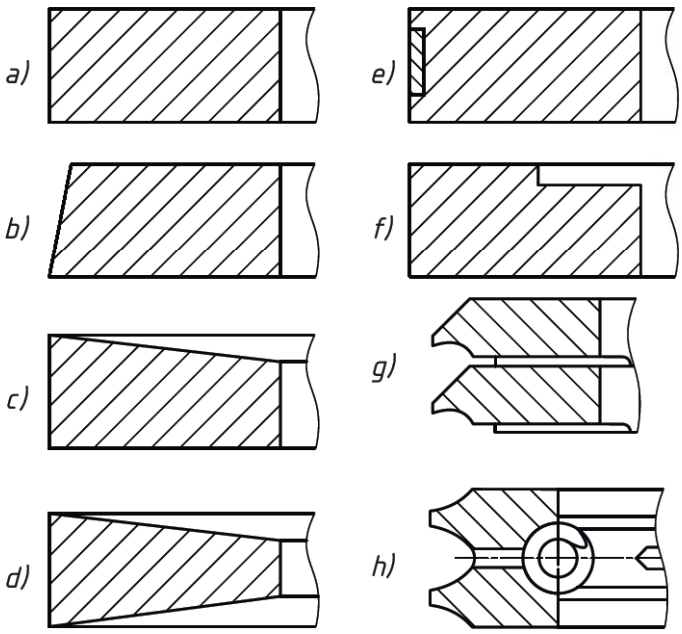


Figure 3.25 – Profile of piston rings in radial section. *a* – rectangular; *b* – trapezoidal; *c*, *d* – cone rings; *e* – rectangular with composite insert; *f* – rectangular twisting; *g* – scraper type oil-removing ring; *h* – box type oil-removing ring.

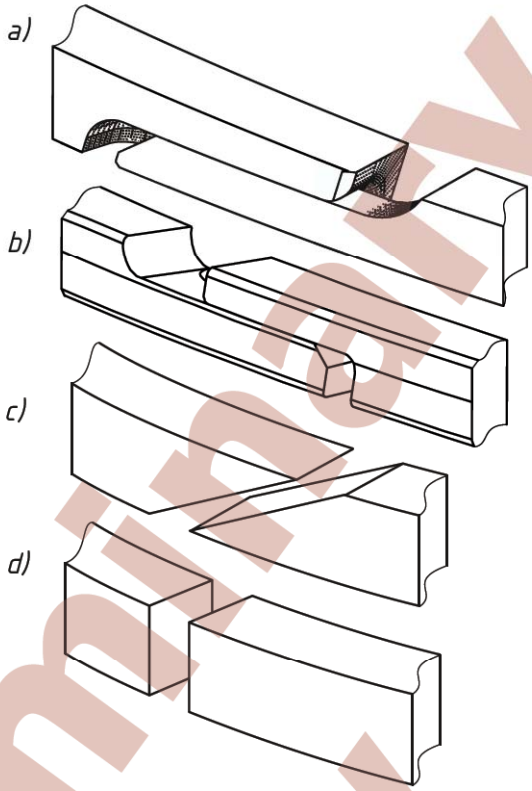


Figure 3.26 – Types of locks in piston rings: *a* – stepped profiled; *b* – stepped overhead; *c* – with inclined cut; *d* – with straight cut

The piston ring system is not an absolutely airtight seal, so gases pass into the gaps between the cylindrical surface of the piston and the sleeve, gaps in the caps, and also into the locking gaps of the rings. The ring locks are positioned relative to each other by shifting them to a certain angle, resulting in a labyrinth seal. Approximate distribution of pressures in the labyrinth seal system is presented in Fig. 3.27. As can be seen from the figure the maximum pressure acts on the first ring p_1, which is about 75...80% of the pressure in the cylinder p_{cyl}. On the subsequent rings the pressure is significantly reduced due to throttling in the gaps.

Compression rings of composite type are used in low speed engines with aluminium or bronze ring inserts (Fig. 3.25 *e*) or special coatings (Fig. 3.28) rolled into the working surface. This allows to reduce the friction forces of the ring against the bushing to reduce the wear of both sealing and end faces and accelerate their running-in. To increase wear resistance and friction coefficient of compression rings, special coatings applied to the working surfaces, porous chromium plating, as well as spraying of carbide-chromium-molybdenum layer are used. The layer of porous chromium, saturated with oil, helps to reduce friction forces, and the molybdenum layer helps to increase the anti-seize properties of the ring. The chrome plating of the lower end surface makes it hard and smooth, which prevents the entrapment of abrasive particles, thus reducing the wear of the ring groove.

The passage of gases through the ring's locking gap leads to a significant temperature increase at the joint, and the lubricating oil experiences a high thermal load, which leads to deterioration of lubrication conditions, up to the rupture of the oil film due to its deflation. Increased local temperatures lead to uneven thermal expansion of the ring and high surface pressures between the liner and the ring. In addition, the gas jet from the upper locking gap leads to localized thermal overloading of the next piston ring.

In order to prevent the above-mentioned negative factors, a ring with a so-called gas-tight joint is used as the upper ring in the low speed engine (Fig. 3.26 *a*). These rings provide good sealing, which in turn leads to violation of pressure distribution on the rings installed below. To eliminate this defect on the working surface of the upper ring is made from four to six inclined grooves

For preliminary review

(Fig. 3.29 *a*, *b*) controlled leakage (CL) providing a distributed flow of gases to the lower piston rings. As the piston rings wear, the depth of the grooves decreases, which leads to a reduction in gas flow, especially significant in engines with relatively small cylinder diameters. Therefore, for engines with a cylinder diameter of less than 50 cm developed a design of top piston rings with end grooves of pressure distribution (Fig. 3.30 *a*, *b*) called CPR-POP (Port-On-Plane). They are made on the underside of the piston ring and have a wedge shape, resulting in an increasing cross-sectional area as the ring wears. So for every 1 mm of radial wear, the clearance increases by 2 mm, resulting in a smaller pressure drop in the channel. By the depth of the CL grooves or the width of the end grooves, it is possible to monitor the wear of the piston rings through the purge ports during preventive inspections.

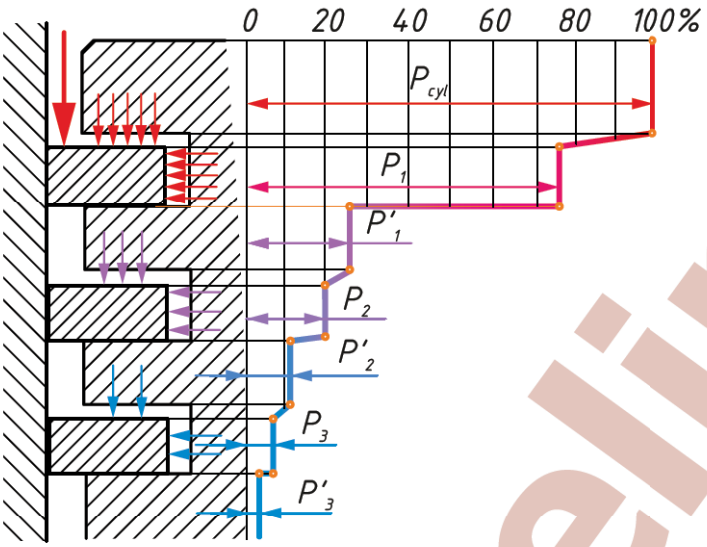


Figure 3.27 – Pressure distribution in a labyrinth seal formed by a system of piston compression rings

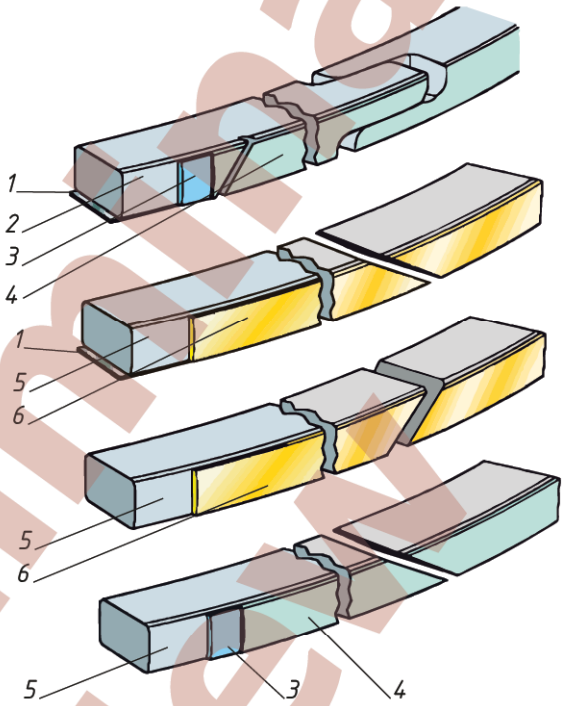


Figure 3.28 – Compression rings of composite type of low-speed diesel engines: 1 – chrome-plated bottom surface; 2 – vermicular cast iron (base material); 3 – hard coating PM2 chrome-carbide 0.5 mm; 4 – Ni-Gr soft prework coating 0.1 mm; 5 – base material of gray cast iron RM5; 6 – aluminium coating (adapted from [35])

The profile of the ring in the radial section significantly affects the efficiency of its operation. Rings with rectangular profile are used in engines with moderate forcing of the operating process (Fig. 3.25 *a*). In the case of boosted Medium Speed Engine and High Speed Engine, rings with trapezoidal profile are used as the upper ones as they are less prone to coking (Fig. 3.25 *c*, *d*). As the bottom ones the so-called «minute» rings with the angle of inclination of the outer forming surface 60...90' are used. Improvement of sealing is achieved by application of rings with asymmetrical cross-section (Fig. 3.25 *f*), providing twisting of the ring during operation, which reduces its tendency to hang-up.

In addition to their sealing function, the rings play an important role in dissipating heat from the piston. In engines with uncooled pistons, up to 50% of the heat absorbed by the piston is dissipated through the rings. For more intensive heat dissipation from the piston rings, they are located as low as possible on the head, so that at TDC they are at the level of the sleeve cooling jacket cavity.

Based on operating conditions, the material of piston rings must have sufficient strength at high temperatures, wear resistance, low coefficient of friction in conditions of limited lubrication and high temperatures. The basic material for rings is gray pearlite cast iron alloyed with molybdenum and vanadium, high-strength cast iron with vermicular graphite, as well as alloy steels.

To remove excess oil from the cylinder sleeve surface in four-stroke engines, in addition to com-

For preliminary review

pression rings, special profile rings called oil rings (Fig. 3.25 *g*, *h*) are installed. By their design, the oil rings are of the scraper (Fig. 3.25 *g*) and box (Fig. 3.25 *h*) type. Usually one oil ring is installed on the piston immediately below the compression set, but on some engines two rings may be installed. The second ring is usually installed in a groove on the bottom of the trunk. Oil-removal rings have increased radial pressure on the cylinder wall, which is increased by using expanders in the form of flat profiled or cylindrical coiled springs. The total pressure on the cylinder wall reaches 3.0 MPa. For drainage of oil removed from the walls, around the perimeter of the cap grind slots that through a system of radial holes are connected to the piston space. Oil rings are made of the same materials as compression rings.

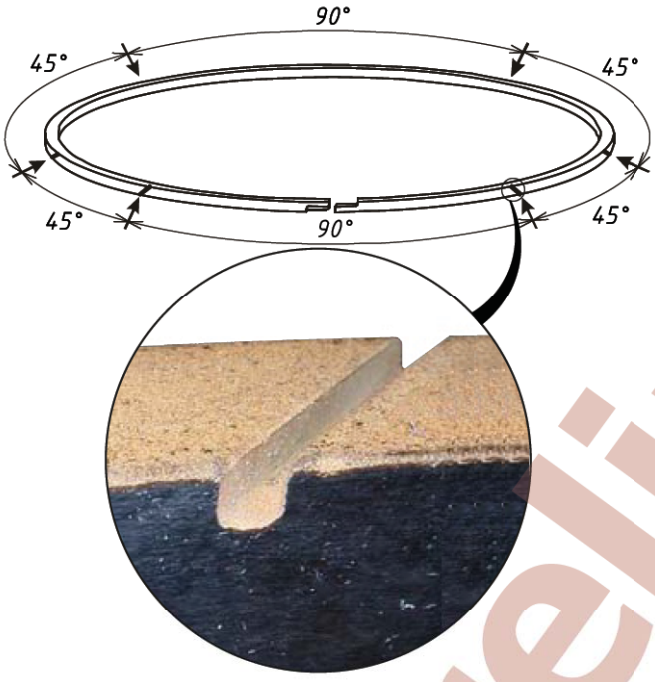


Figure 3.29 – Location of controlled leakage grooves (*a*) and their general view on the working surface of the ring (*b*) (adapted from [36])

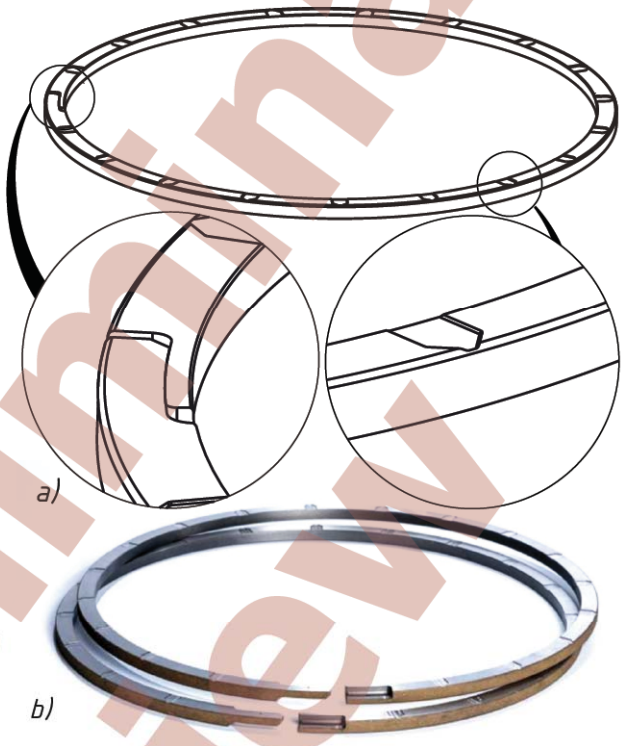


Figure 3.30 – Location of end grooves of controlled leakage (*a*) and general view of rings with this type of grooves (*b*) (adapted from [37])

3.2.4 Piston rod and rod packing

The piston rod connects the piston to the crosshead crossmember and transmits forces from the piston to the crosshead and vice versa. The piston rod of modern low speed engines is a hollow cylinder with two flanges at the ends. To the upper flange long pliable studs or bolts are attached piston elements (Fig. 3.14). Sometimes spacers are used to increase the length of studs. The lower flange connects the rod to the crosshead crossmember by means of studs or bolts, which are screwed into the crossmember through holes in the rod flange.

A flat gasket (Fig. 3.14 *c*) is installed between the end cutout in the crosshead crosshead and the flange, its thickness is used to adjust the initial position of the piston relative to the cylinder sleeve.

In modern low speed engines, the piston rod has a through-hole in which a pipe is installed to supply and discharge coolant to the piston. In the RT-flex and X series engines from Wärtsilä and WinGD, the coolant is fed into the gap between the pipe and the piston rod and discharged through the inside of the pipe, thus reducing the rod temperature and improving the conditions for the oil seal. In MAN engines of K, L, S, G series, coolant is supplied to the piston through the pipe cavity, and discharged into the gap between the pipe and the rod (Fig. 3.31). The outer surface of the rod is ground to fit effectively against the rod packing seals and to reduce friction forces.

For preliminary review

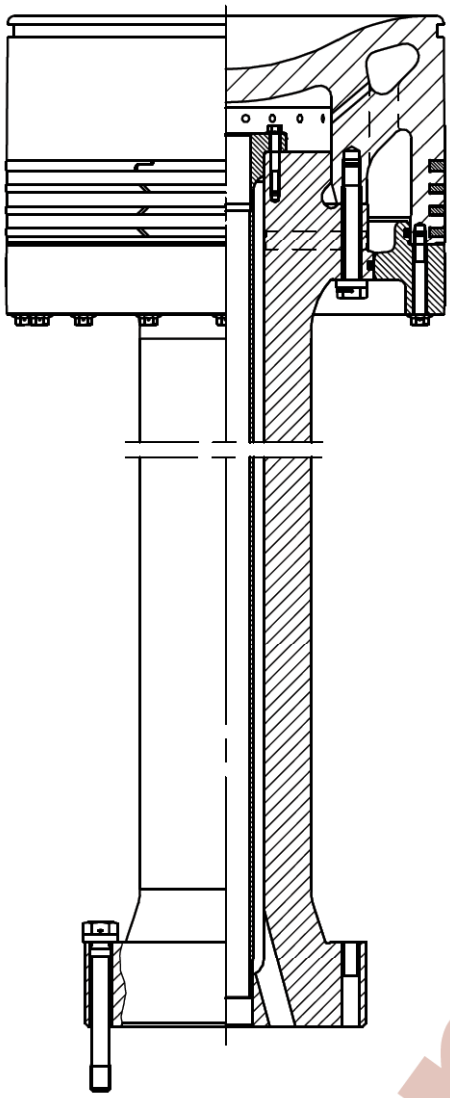


Figure 3.31 – MAN low speed engine piston and piston rod (adapted from [25])

At the point of rod passage through the diaphragm separating the cylinder piston cavity from the crankcase, an oil seal is installed (Fig. 3.32) preventing lubricating oil from the crankcase to the purge air cavity and purge air to the crankcase. The housing of the gland is made in the form of a flange consisting of two parts connected by bolts. On the inner side of the flange there is a number of annular grooves. In the upper groove is installed oil ring, which prevents the ingress of sludge from the piston cavity to the other rings. On the crankcase side, in the lower grooves, oil rings are installed to remove lubricating oil from the rod. From the lower grooves, the oil removed from the rod by the rings is returned to the crankcase through holes in the stuffing box housing. O-rings are located between the oil rings to prevent blow-by air from penetrating along the piston rod. Structurally, the O-rings and O-rings are a set of three or four segments that are held in place by locating pins. The segments are pressed against the piston rod by means of coupling springs located in grooves cut on the outer surface of the ring. On the inner surface of the oil sealing rings there are oil sealing lips. The sealing rings have a flat working surface. In one groove, the segments are placed in two rows with the locks offset relative to each other. The gaps between the ends of the segments ensure their fit to the piston rod as the segment wears.

Through the holes in the housing and the drilling in the diaphragm, the groove of the upper oil ring communicates with a funnel outside the engine, which is used to check the condition of the o-rings and oil rings. A burst of air indicates defects in the o-rings, and excess oil indicates defects in the oil rings. The oil seal is removed with the piston rod during inspection, but can also be disassembled in the crankcase without removing the piston.

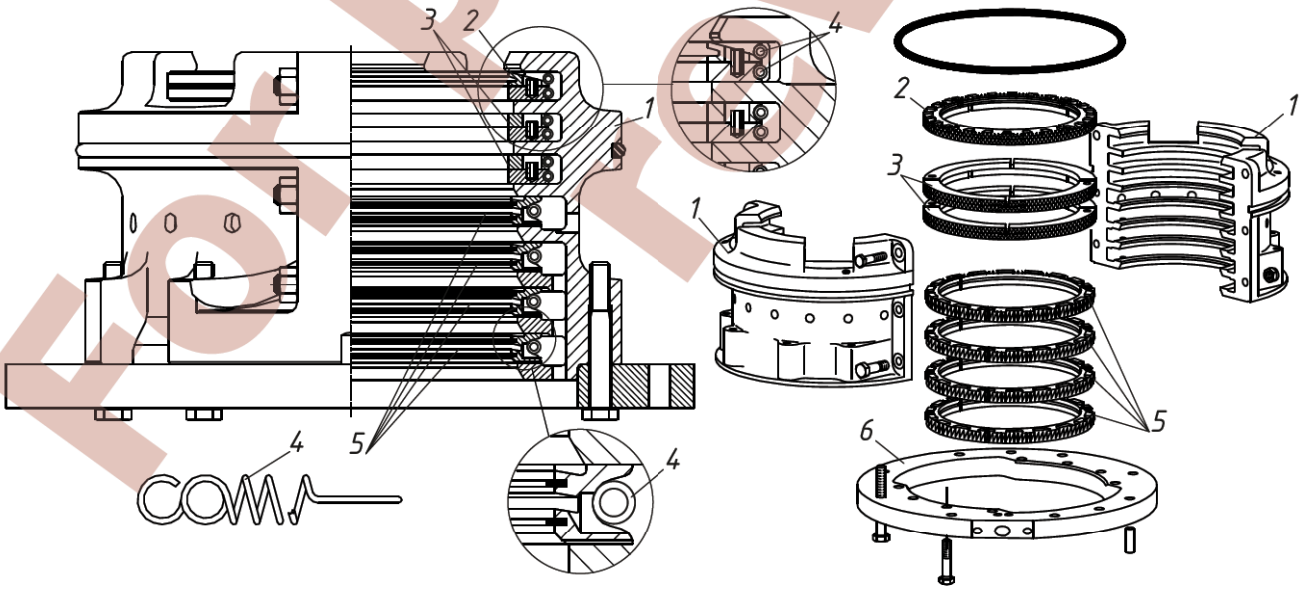


Figure 3.32 – Piston rod oil seal of MAN K, L, S, G series engines. 1 – gland housing; 2 – upper oil ring; 3 – sealing rings; 4 – coupling springs; 5 – lower oil rings; 6 – base of the gland housing (adapted from [21])

For preliminary review

3.2.5 Crosshead

The crosshead absorbs the forces from gas pressure transmitted from the piston through the piston rod and also transmits the normal force from the connecting rod to the parallels, thus relieving the piston rod and the piston itself from lateral forces. The crossheads are designed as single-sided or double-sided slider. One-sided slider is used on non-reversible engines, as well as on reversible engines, where parallels are made in the form of grooves located on both sides of the slider. The disadvantage of this scheme is that when the engine is reversed, the area that absorbs the normal force is smaller than when the diesel engine rotates to the front. As a result, it is necessary to limit the power of the engine when it operates in reverse mode. Modern low speed engines use mainly double-acting crosshead mechanisms, which provide the possibility of obtaining maximum power regardless of the direction of engine rotation. The general layout of a double-acting crosshead crank is shown In Fig. 3.33 *a* and a general view of the crosshead assembly is shown In Fig. 3.33 *b* and 3.33 *e*.

The crosshead consists of a cross member and sliders.

The crossmember transmits forces from the piston to the connecting rod by connecting the piston rod and sliders to the top head of the connecting rod in a single assembly called the main connection. The middle portion of the crossmember has a butt face for attaching the piston rod flange. The use of a flange connection allows for increased crossmember stiffness and the use of a crosshead bearing with a solid bottom surface. This makes it possible to reduce the specific pressures on the bearing surface, thereby increasing bearing life.

Threaded holes for mounting bolts, holes for locating pins, as well as channels for coolant inlet and outlet (Fig. 3.33 *d*) are made on the butt face of the crossmember. At the edges of the butt face are journals for connecting to the trunnions of the upper connecting rod head. At the edges of the crossmember, two sliders are mounted on special journals on either side of the connecting rod swing plane. In addition, the oil system pipes are attached to the machined surfaces of the crossmember.

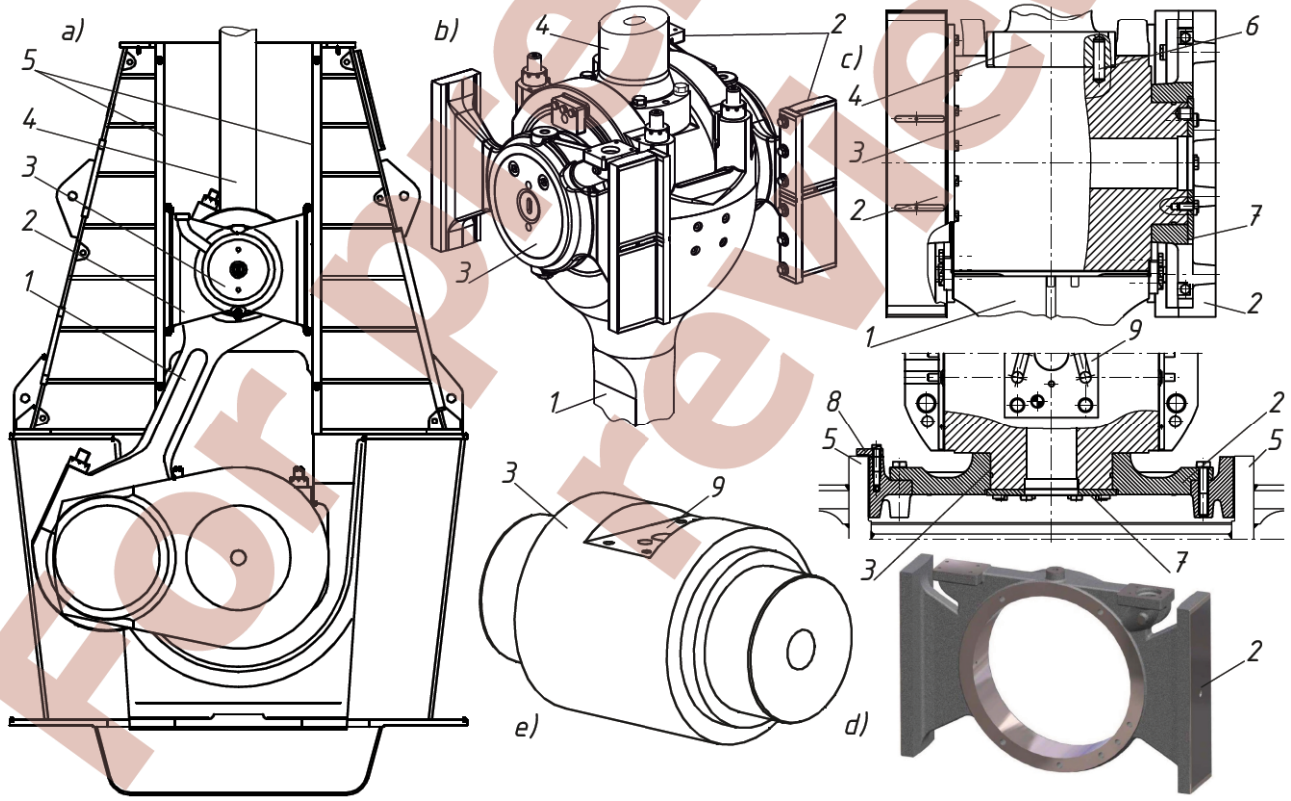


Figure 3.33 – Crosshead and its elements: *a* – crosshead assembly location in the engine crankcase; *b* – general view of the double-sided crosshead assembly of a slow-speed engine; *c* – crosshead assembly of K, L, S series engines of MAN; *d* – general view of the double-sided slider; *e* – general view of the crosshead crossmember; 1 – connecting rod; 2 – slider; 3 – crosshead; 4 – piston rod; 5 – parallels; 6 – locating pin; 7 – fixing covers; 8 – side support strips; 9 – crosshead buttress surface (adapted from [21, 25])

For preliminary review

The double-sided arrangement of the sliders provides easy access to the main coupling bearing, makes it possible to reduce the length of connecting rods and reinforce the bed posts with slider guides to increase the rigidity of the engine frame. The crossmembers are manufactured using carbon, chromium-molybdenum and chromium-nickel medium-carbon heat-treated steels.

Sliders move along parallels in the connecting rod swing plane and are made solid (Fig. 3.33 *b*, *d*) or composite (Fig. 3.33 *c*). They have two working surfaces filled with antifriction alloy, on which longitudinal and transverse grooves are made for uniform distribution of oil over the entire plane. To fix the sliders from displacement in the longitudinal plane of the engine, side support strips 8 are installed (Fig. 3.33 *c*). The sliders are cast from carbon steel.

Parallels absorb the lateral force transmitted by the crosshead sliders. They are vertical cast iron or steel guides, which are attached to the cross plates of the bed. Parallels can be single or double-sided. Single-sided parallels are used in non-reversible engines, and in reversible parallels are equipped with removable cheeks that form a double-sided groove on which the slider slides. Double-sided parallels have four guiding surfaces that work in pairs. They are mounted on either side of the connecting rod swing plane with an offset from the cylinder axis of approximately half the width of the cross member.

3.2.6 Connecting rods

The connecting rod group includes a connecting rod, bearings of the upper and lower heads, connecting rod caps, connecting rod bolts and nuts (Fig. 3.34). The connecting rod transmits forces from the piston pin or crosshead to the crankpin of the crankshaft, and also ensures the movement of the piston during auxiliary strokes. When the engine is running, the connecting rod undergoes a complex translational-rotational motion, in which it is acted upon by forces from gas pressure, inertial forces of translationally and rotationally moving masses, and inertial forces that arise when the connecting rod swings.

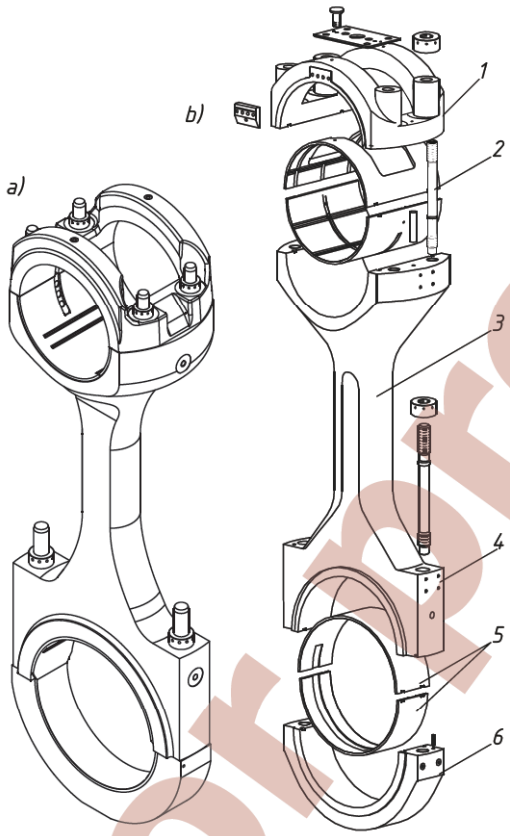


Figure 3.34 – Connecting rods of modern crosshead engines: *a* – RT-flex series from Wärtsilä; *b* – MC series from MAN: 1 – upper head cover; 2 – cross member bearing; 3 – rod; 4 – lower head; 5 – lower head bearing; 6 – lower head cover (adapted from [17, 19])

The connecting rod consists of upper and lower heads, which are connected to each other by a rod with smooth transitions from one structural element to another (Fig. 3.34, 3.35). One or more rods may be attached to the lower head of the connecting rod. If there are several rods, then one of them is rigidly attached to the head, and the rest are hinged through special eyes into which connecting fingers are inserted (Fig. 3.35 *g*). In this design, the rigidly connected connecting rod is called the main connecting rod, and the hinged connecting rod is called the trailing connecting rod. A similar connecting rod arrangement is typical for some high-speed V-shaped engines. It allows you to reduce the length of the connecting rod journals of the shaft, thereby increasing their rigidity, as well as more evenly distribute the specific loads on the bearing of the lower head of the connecting rod. The disadvantage of this design is that the kinematics are more complex, as a result of which the pistons driven by the trailing connecting rod have a shorter stroke than the pistons driven by the main connecting rod. In addition, such connecting rods are structurally more complex, and the presence of a hinge joint, which experiences heavy loads during operation, reduces the reliability of the entire assembly. For these reasons, in V-shaped marine MSEs, a scheme is predominantly used in which

For preliminary review

two independent connecting rods are attached to one crankpin of the shaft, from which the pistons of the left and right rows of cylinders are driven.

The connecting rod experiences large axial compression and tension forces from the action of the components of gas pressure forces and inertia forces of forward moving masses. To ensure the necessary rigidity, the connecting rod for LSE and MSE is made predominantly of a circular cross-section (Fig. 3.34; 3.35 *a*, *d*, *e*, *f*; 3.36 *a*) with drilling for supplying oil to the upper head of the connecting rod or without drilling. In HSE, to reduce the mass of the connecting rod, I-beam rods with and without drilling are widely used (Fig. 3.35 *b*, *c*, *g*; 3.36 *b*, *c*). This shape allows for a smooth transition of the side shelves into the curvilinear outlines of the heads.

The connecting rod heads are made solid or split. The detachable part of the head is called the cap. In modern crosshead engines, to facilitate installation and dismantling, both heads are made detachable (Fig. 3.34). For four-stroke diesel engines, only the lower head is detachable, which is attached to the connecting rod housing with studs or bolts.

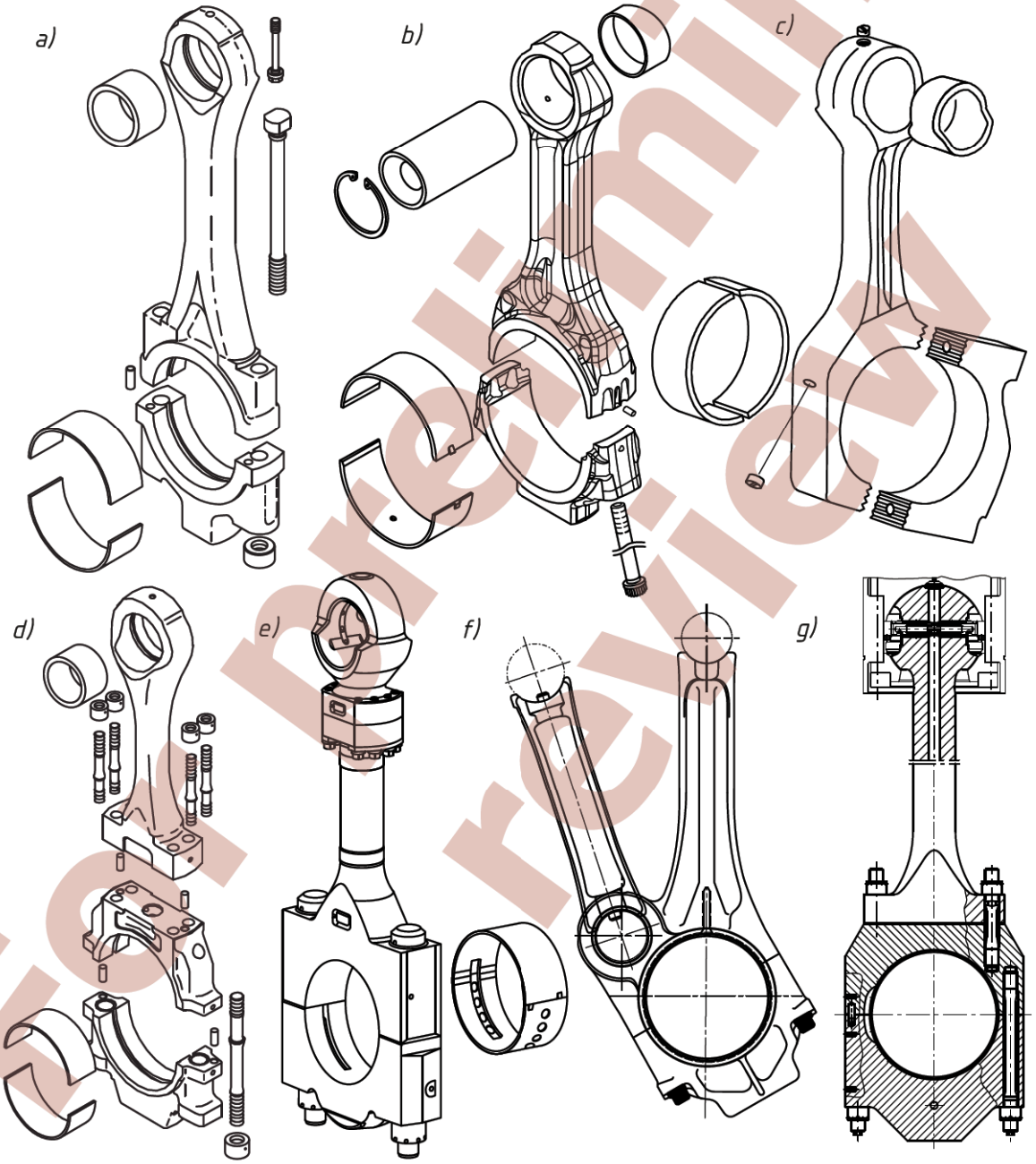


Figure 3.35 – Basic designs of connecting rods for medium- and high-speed marine internal combustion engines. *a* – L 16/24 from MAN; *b* – Cat 3500 from Caterpillar; *c* – L 23/30 from MAN; *d* – L27/38 from MAN; *e* – M 43 from MaK; *f* – 7FDS from General Electric; *g* – Z40 from Sulzer with spherical support (adapted from [12, 15, 29, 30, 34, 38, 39])

For preliminary review

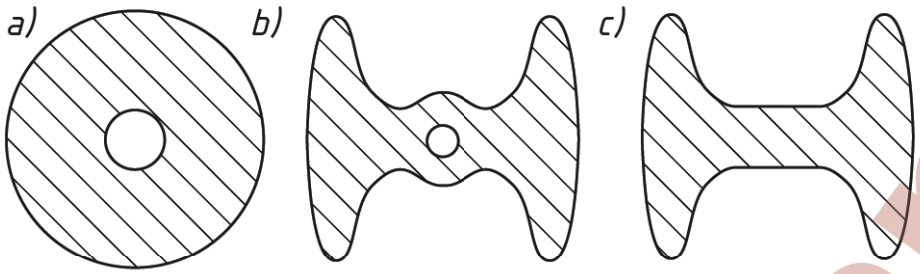


Figure 3.36 – Types of main sections of the connecting rod of marine engines: *a* – round with drilling for supplying oil to the upper head; *b* – I-beam with drilling; *c* – I-beam without drilling

The upper head of the connecting rod of a modern crosshead engine is made in the form of a split eye, which is a smooth continuation of the rod. The eye has a bore for the cross member bearing. The top cover has a cutout for the passage of the piston rod. To fix the position of the cover relative to the head housing, locating pins are used. The cross member bearing is made in the form of two liners, filled with a layer of white metal, and the lower liner is also covered with a running-in layer. For a more uniform supply of oil to the entire surface of the bearing, the liners have oil distribution grooves and thinning areas called pockets. They allow you to increase oil flow through the bearing, thereby improving the conditions for cooling the liners, as well as for removing wear products from the friction zone (Fig. 3.37).

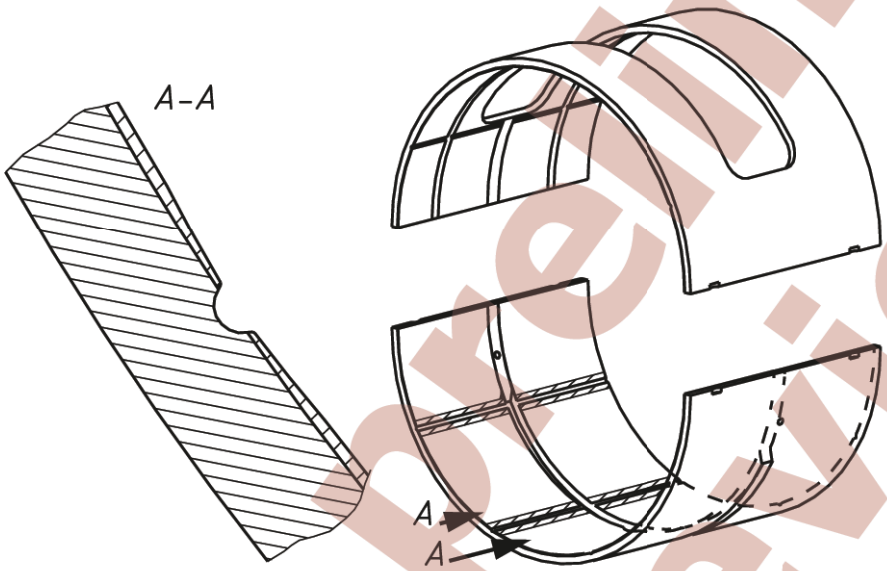


Figure 3.37 – Design of the LSE cross member bearing of the MC series from MAN (adapted from [21])

The upper head of the connecting rods MSE and HSE is usually made in the form of a solid closed eye, which has a smooth interface with the connecting rod and is symmetrical relative to its longitudinal axis. In some cases, the upper head is made with oblique cuts or profile recesses of the end surfaces (Fig. 3.35 *a*, *b*, *d*, *e*), which allows, without reducing the area of the most loaded lower part of the piston pin bearing, to reduce the total mass of the connecting rod. For engines with forced cooling of the pistons, the upper part of the head is made in the form of a cylindrical or spherical surface, to which a special spring-loaded cup is tightly pressed to supply oil from the connecting rod to the piston.

In a number of engines, the upper head can be made in the form of a spherical support (Fig. 3.35 *f*). This design allows you to more evenly distribute the loads over the surface of the bearing, and also, using a special mechanism located in the support, rotate the piston during operation (Fig. 3.16 *e*).

To facilitate piston removal, some engines have a removable upper head. The head is attached to the rod via a flange connection using bolts (Fig. 3.35 *e*).

To reduce friction, bearings in the form of bushings made of bronze or steel with an anti-friction layer are installed in the upper head. To ensure uniform distribution of oil, as well as supply oil to

For preliminary review

the piston, a system of grooves and holes is made in the bushing, the location of which depends on the design features of the upper head.

In engines with a rigidly mounted piston pin, the upper head of the connecting rod is made in the form of a flange with a bed for laying the latter (Fig. 3.35 *g*). The pin is secured using elongated fitting bolts (Fig. 3.21).

The lower head of the connecting rod is an eye with a bore for the connecting rod bearing. According to the conditions for assembling the connecting rod with the crankshaft, the lower head of the connecting rod is split. The connector can be made both in a plane perpendicular to the axis of the connecting rod, and at an angle. An inclined connector makes it possible to reduce the bore size of the connecting rod with the cover removed, which makes it possible to insert the lower head of the connecting rod of increased diameter through a cylinder sleeve of a relatively small diameter (Fig. 3.38). The disadvantages of this design include the difficulties associated with reversing engines with such a connecting rod, since when the direction of rotation changes, the bolts securing the bottom cover will absorb significant tensile forces. The inclined connector is also used for more convenient removal of the cover through the side hatches of the crankcase.

It is extremely rare that in order to reduce the passage dimensions of the lower head of the connecting rod, it is made consisting of three parts. In Fig. 3.39 *a* shows the connecting rod design of the so-called fork design used on Electro-Motive Diesel EMD 710 engines. In this design, the lower bearing housing consists of three parts of the connecting rod and two semi-housings, the dismantling of which makes it possible to significantly reduce the passage dimensions of the lower head. The latter is clearly visible in Fig. 3.39 *b*, which shows the piston kit for this engine.

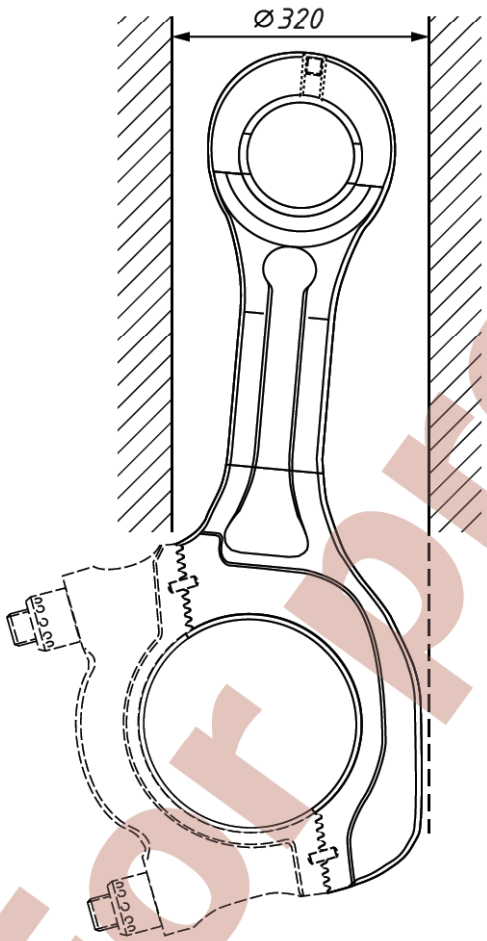


Figure 3.38 – Installation of a connecting rod for a Vasa R 32 engine from Wärtsilä with an oblique connector through a bushing of limited diameter (adapted from [40])

Precise fixation of the position of the covers relative to the connecting rod housing is carried out using locating pins, a special toothed notch or tight-fitting bolts.

In a number of designs of medium-speed engines, connecting rods of the so-called marine type are used, in which the lower head is removable. It is connected to the rod through a flange (Fig. 3.35 *d*, *f*). This allows you to change the connecting rod journal bearing without removing the piston, and also remove the piston without disassembling the connecting rod bearing. A gasket is installed between the rod flange and the lower head, which regulates the initial position of the piston relative to the cylinder liner. This allows you to compensate for some inaccuracies in the connections when assembling the engine, as well as adjust the geometric compression ratio. For example, on a Wärtsilä 46 series engine, installing a thick gasket 11.04 mm provides a compression ratio of 14:1, characteristic of the first modifications of engines in this series. During the engine upgrade process, the compression ratio was increased to 16:1 by installing a thick gasket 17 mm.

Connecting rods are made from carbon or alloy steel by forging followed by machining.

The individual parts of the connecting rods are connected to each other using bolts or studs with nuts. These connecting rod design elements are extremely important, since their breakage often leads to failure of such expensive engine elements as the crankcase and crankshaft. The studs are installed perpendicular to the plane of the bottom head connector and additionally serve as guides when installing the bottom cover. Using studs requires

For preliminary review

more space to mount the lower bearing cap, so more compact engine designs use bolted connections. In this case, the bolts can be installed either through, tightened with nuts (Fig. 3.35 *a*, *e*), or dead-end, screwed into threaded holes in the connecting rod housing (Fig. 3.35 *b*, *c*, *g*). In the first case, the bolt head is given a special shape that prevents it from turning (Fig. 3.40 *b*). The bolts are usually installed perpendicular to the parting plane, but in some cases, for a more uniform distribution of forces, they are installed at a certain angle (Fig. 3.35 *b*).

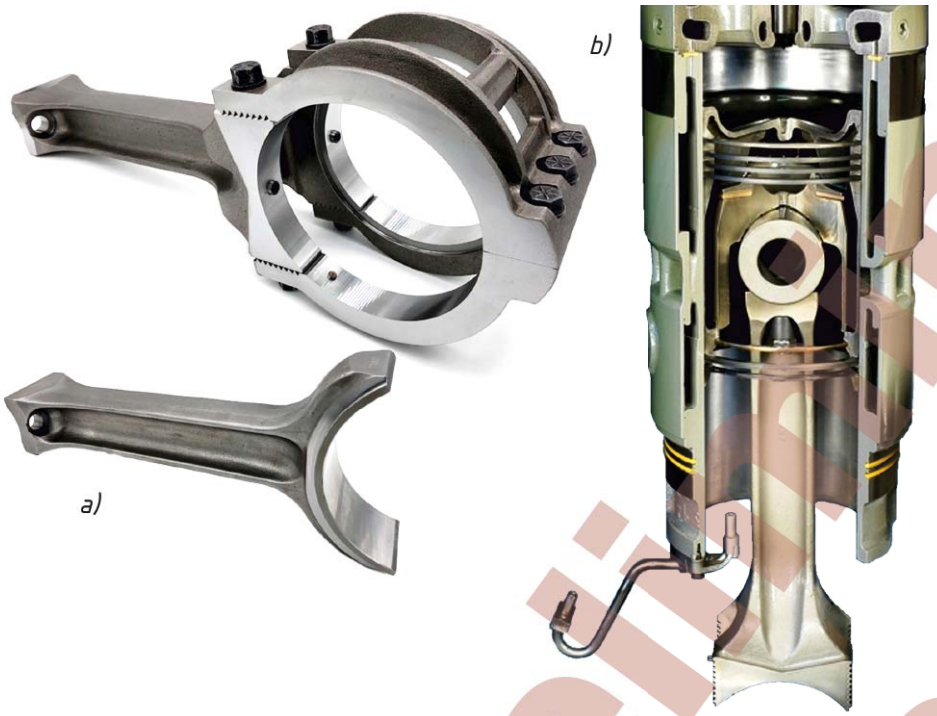


Figure 3.39 – Design of the forked connecting rod of the EMD 710 engine from Electro-Motive Diesel (*a*) and general view of the cylinder-piston kit (*b*) (adapted from [41, 42])

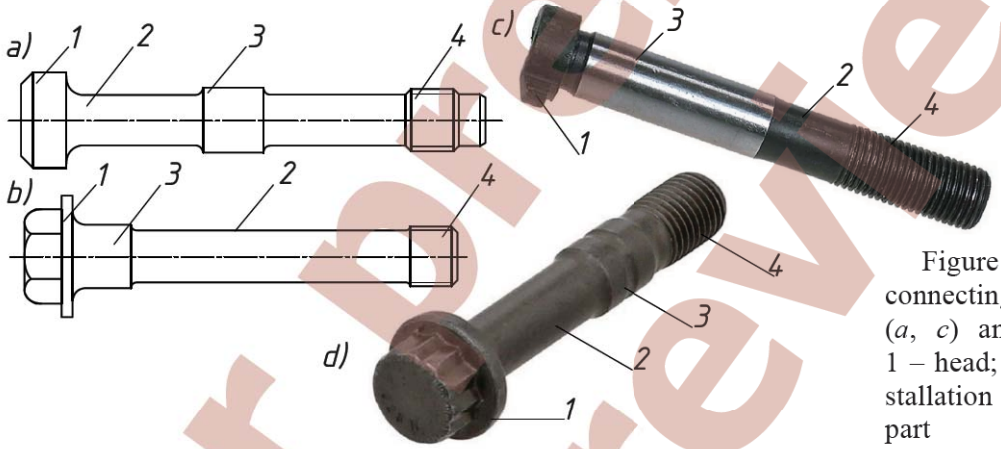


Figure 3.40 – Design of connecting rod bolts, through (*a*, *c*) and dead-end (*b*, *d*): 1 – head; 2 – housing; 3 – installation belt; 4 – threaded part

An important point when assembling connecting rods is their correct tightening, which is usually done using torque wrenches or hydraulic jacks. Insufficient tightening leads to the fact that during operation, under the influence of alternating forces, a gap periodically forms between the joints, leading to the occurrence of shock loads and the formation of hardening. Over-tightening can lead to loosening of threaded connections due to plastic deformation of the bolt material or even to its breakage. In addition, it leads to deformation of the parts being tightened. In both cases, the original geometry of the connecting rod is violated, and the operating conditions of its bearings worsen.

In some connecting rod designs, the bolts act as guide elements. In this case, installation belts are made on them, which fit tightly into calibrated holes on the mating parts. To ensure the necessary strength, connecting rod bolts and studs are made from high-quality alloy steels, and the transitions between their individual elements are made as large as possible. The threaded part of the bolt is made by knurling with a large radius of curvature at the tops and bottoms, and the remaining surfaces are carefully ground. When disassembling the bearings of the lower head of the connecting rod, the bolts or studs are checked for lengthening or decreasing the diameter in the narrowest sec-

For preliminary review

tion. If there are extreme deviations, the studs or bolts are rejected and replaced with new ones.

To assemble connecting rods, castle or special cylindrical nuts with radial drillings are most often used (Fig. 3.38). Typically, to relieve the first most loaded threads, the bearing surface of the nuts is made below the last thread of the bolt or stud. Special nuts, as a rule, are self-locking, and castle nuts are locked with cotter pins.

The sliding bearings of the lower head of the connecting rod are made in the form of two thin-walled liners (Fig. 3.41) made of steel and coated with several layers of antifriction material, which can be from 2 to 5 (Fig. 3.42).

The liners are secured against rotation using locating pins or special bends called locks, which fit into milled recesses on the bearing bed. For a more uniform distribution of oil over the friction surface, annular grooves are made on the lower liners, and special thinnings called pockets are made in the area of the joints (Fig. 3.41). They allow you to increase oil flow through the bearing, thereby improving the conditions for cooling the bearings, and also serve to remove wear products from the friction zone.

Figure 3.42 shows the design of a five-layer bearing shell of a modern MSE.

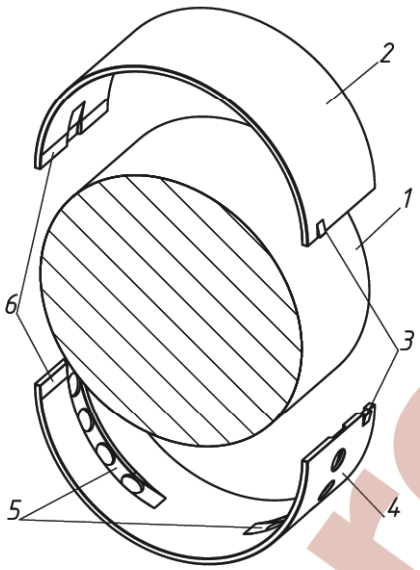


Figure 3.41 – Design of connecting rod bearings for the MaK 43 marine medium-speed engine. 1 – crankshaft connecting rod journal; 2 – upper liner; 3 – lock bend; 4 – lower liner; 5 – annular, oil distribution grooves; 6 – oil pockets (adapted from [32])

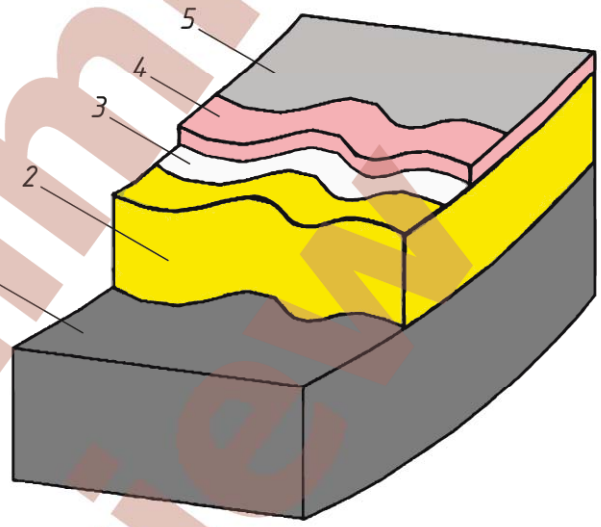


Figure 3.42 – Diagram of the arrangement of layers of the connecting rod bearing of the Wärtsilä 46 series engine. 1 – steel base; 2 – layer of lead bronze; 3 – nickel separating layer; 4 – working layer of babbitt (*AlSn alloy*); 5 – running-in layer of zinc (adapted from [43])

3.2.7 Crankshaft group

The crankshaft group includes the crankshaft itself, the counterweight system, gears or sprockets of the timing drives and auxiliary mechanisms, the torsional vibration damper and the flywheel.

Crankshaft – serves to convert the translational motion of the pistons into rotational motion and to transfer mechanical energy to its consumers. This is one of the most critical, expensive and difficult parts to manufacture. The need to replace or repair the crankshaft is fraught with great difficulties, so the need to carry out such an operation for LSE and some MSEs determines the engine life (before lifting the shaft) which for MSE is 50...100 thousand hours, and for LSE 100...200 thousand hours. Such a resource is quite comparable to the full service life of the vessel itself. For HSE, shaft excavation is not such a difficult operation, so it is performed every 20...50 thousand hours.

Structurally, the crankshaft consists of main (frame) and connecting rod (crank) journals, which are connected using cheeks. The names of the crankshaft elements given in parentheses are more typical for LSEs, which inherited these names from steam engines.

The main journals are located along the axis of rotation of the shaft and rest on frame bearings. The outer journals are made elongated by placing ridges for thrust bearings, mounting surfaces for

For preliminary review

gears or sprockets for driving auxiliary mechanisms, and a flange for attaching the flywheel on the parts protruding from the bearing (Fig. 3.43).

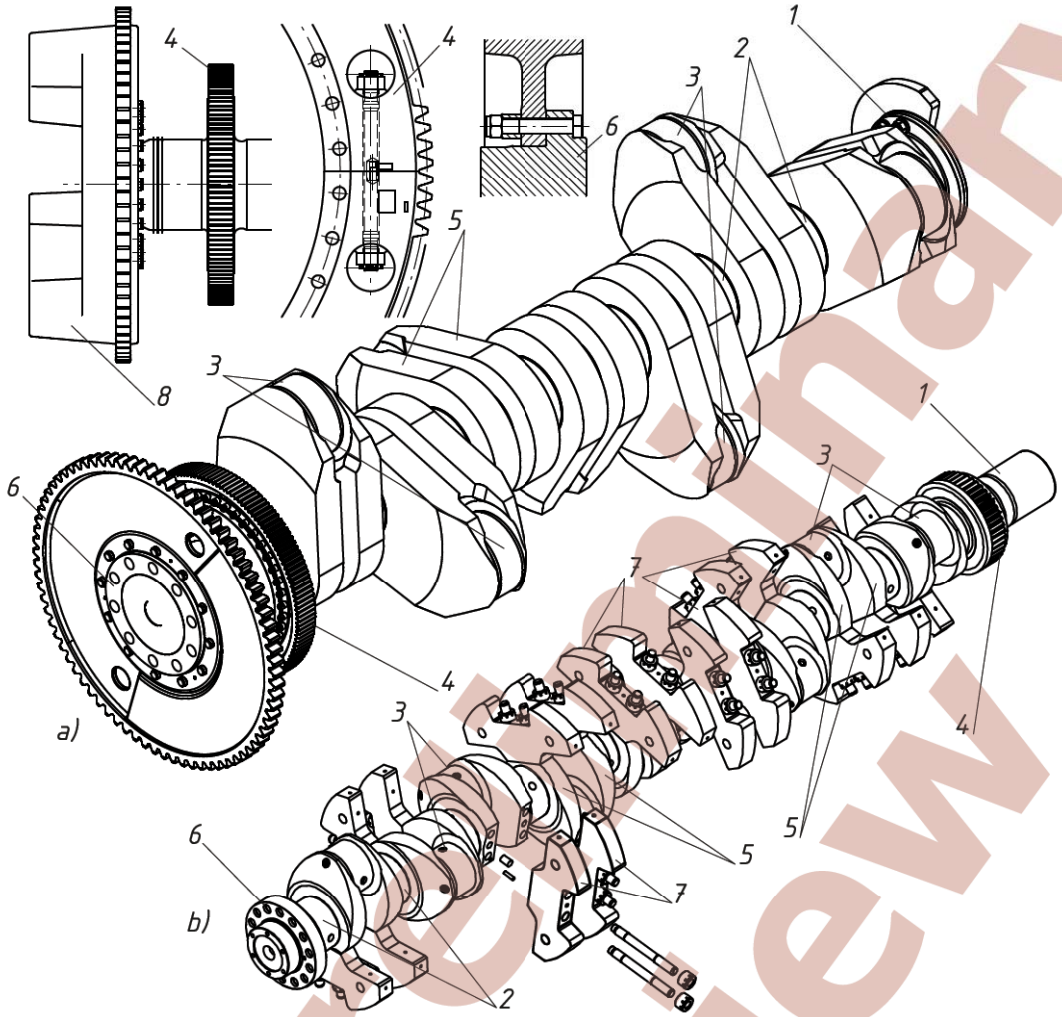


Figure 3.43 – Crankshafts of marine internal combustion engines. *a* – two-stroke low-speed diesel engine RT-flex60C from Wärtsilä; *b* – four-stroke diesel engine L21/31 from MAN; 1 – flange for driving auxiliary mechanisms ; 2 – frame (root) neck; 3 – crank (crank) neck; 4 – sprocket (gear) for driving gas distribution and auxiliary mechanisms; 5 – crankshaft cheek; 6 – flange for mounting the flywheel; 7 – counterweights; 8 – flywheel (adapted from [44, 45])

LSE crankshafts with a number of cylinders from four to eight are made in the form of one section. With a larger number of cylinders, the shaft is made of two sections connected to each other using flanges. In this case, the sprockets or gears that drive the auxiliary mechanisms and the camshaft are placed in the central part of the engine.

Crankshafts of modern medium- and high-speed engines are usually made in one piece from a billet obtained by forging or casting.

The connecting rod journals in the longitudinal plane are displaced from the axis of rotation of the crankshaft by an amount equal to half the piston stroke, which is called the crank radius. In the plane of rotation, the connecting rod journals are offset relative to each other by an angle called the crank wedge angle φ_{wa}.

For two-stroke engines this angle is usually equal to:

$$\varphi_{wa}=360/i,$$

for four-stroke:

$$\varphi_{wa}=720/i,$$

where i is the number of engine cylinders.

In some cases, to ensure high engine balance, other crank wedge angles are used, and in some

For preliminary review

V-shaped engines, flashes may occur at unequal angular intervals.

In LSE, the main and connecting rod journals usually have the same diameter. For MSE and HSE, to ensure the possibility of dismantling the connecting rod through the cylinder liner, in some cases the diameter of the connecting rod journal is reduced in relation to the diameter of the main journal.

To avoid overloading the main bearings, when choosing the operating order of the cylinders, try to avoid sequential flashes in adjacent cylinders.

When strength conditions allow, reducing the mass of the connecting rod journals, they are made hollow. The internal cavity is often performed eccentrically, shifting the drilling axis upward relative to the axis of the neck itself, by an amount *a* (Fig. 3.44 *d*). This allows the center of gravity (*CG*) of the journal to be shifted closer to the axis of rotation, thereby reducing the centrifugal force it creates. Considering that steel, and especially cast iron, work much better in compression than in tension, the eccentric location of the cavity allows, at the moment of greatest load on the neck, tensile forces to be distributed over a larger cross-sectional area, and compressive forces over a smaller one. As a result, the stiffness of the neck increases.

The cavity in the neck can be used for additional separation of oil supplied for lubrication of connecting rod bearings (Fig. 3.44 *a*). To do this, plugs are installed at the ends of the drillings, and a tube called an oil well is inserted into the lubrication hole, which enters the internal cavity approximately to the level of the journal axis. When the shaft rotates, centrifugal forces act on mechanical impurities in the oil, which throw them to the periphery, where they settle on the surface of the internal cavity of the neck, and purified oil enters the tube.

The main disadvantage of the scheme discussed above is that when starting the engine, the internal cavities take a long time to fill with oil, and all this time it practically does not reach the connecting rod bearings. For this reason, in a number of designs, oil is directly supplied to the bearing through the internal cavity of the journal through a specially pressed bronze tube (Fig. 3.44 *b*).

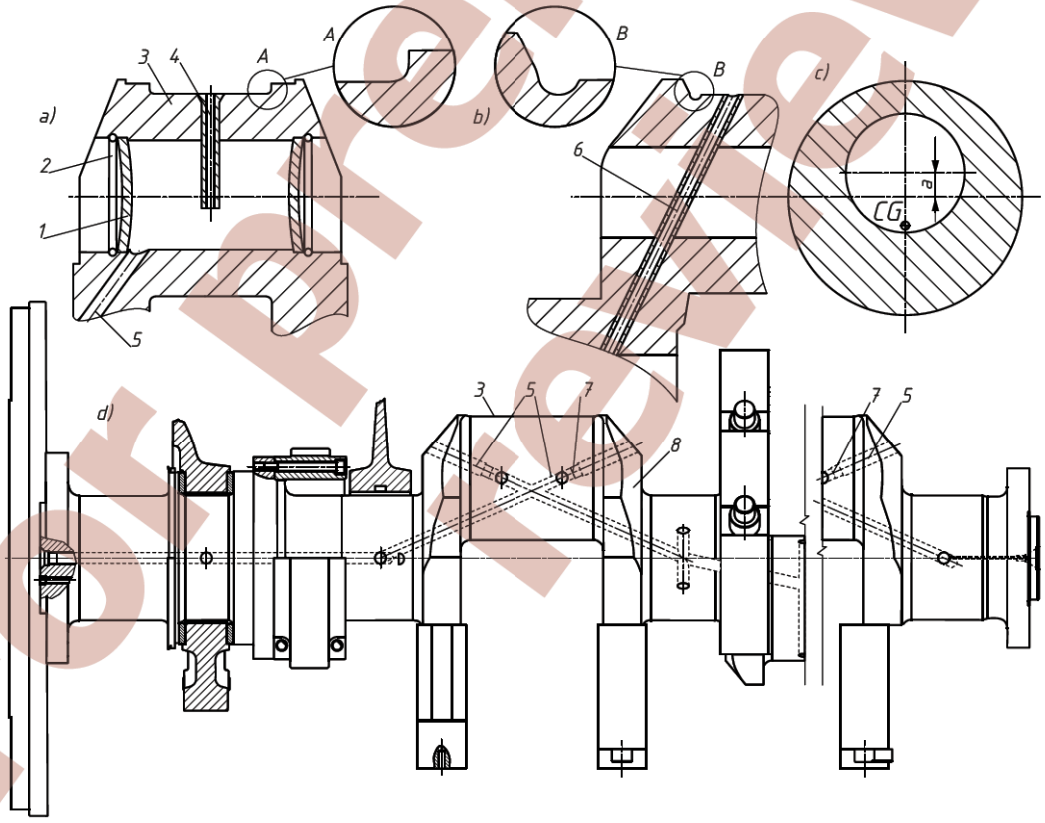


Figure 3.44 – Design of the crankshaft connecting rod journals: *a* – with a cavity for oil separation; *b* – with oil supply through a separate tube; *c* – eccentric location of the internal cavity; *d* – solid neck with inclined drillings. 1 – plugs for the connecting rod journal cavity; 2 – stopper; 3 – connecting rod journal; 4 – oil well tube; 5 – oil supply channel; 6 – oil supply tube; 7 – technological plug of the oil channel; 8 – cheek rounding (adapted from [45])

For preliminary review

On most modern engines with a high level of acceleration of the operating process, based on strength conditions, the crankshaft journals are made solid, providing only drillings for supplying oil to the connecting rod bearings (Fig. 3.44 *c*).

For MSE and HSE, the oil supply to the connecting rod bearings is carried out from the main journals through special inclined and radial drillings. The oil outlet holes are rounded to prevent stress concentration; technological plugs are installed in the remaining holes (Fig. 3.44 *c*).

In LSE, oil for lubrication of the crank bearing is supplied through a drilling in the connecting rod from the oil distribution system of the main connection, where it is supplied from the engine's circulation lubrication system. This solution makes it possible to avoid drilling in the crank necks, which weakens their structure.

The crankshaft cheeks are connected by the main and connecting rod journals. The dimensions and shape of the cheek must provide the necessary strength and rigidity of the crankshaft. The shape of the cheek largely depends on the method of manufacturing the shaft; it can be oval, round, rectangular with rounded edges, etc. In accordance with the load distribution diagram, part of the metal is removed from the cheeks, where this does not affect their rigidity and strength (Fig. 3.43; 3.44 *c*). As a result, they achieve not only a reduction in the weight of the shaft, but also a reduction in the unbalanced masses of the rotating parts.

For engines with moderate long strokes, when choosing journal diameters, try to ensure their overlap (Fig. 3.44 *c*), based on the condition:

$$0.5(d_{cj} + d_{ccj}) > r,$$

where d_{cj}; d_{ccj} – diameter of the main and connecting rod journals, respectively; r – crank radius.

This makes it possible to increase the strength and rigidity of the shaft, or makes it possible to reduce the size of the jaws without reducing the strength and rigidity.

The crankshafts of powerful low-speed marine diesel engines are usually made of composite or semi-composite. The individual parts of such a shaft are connected to each other using a hot or cold press fit with an interference fit. Sometimes the parts are welded together. The individual elements of a semi-compound crankshaft of a modern low-speed engine are shown in Fig. 3.45.

The main element of such a shaft is a crank, consisting of two cheeks connected by a cylindrical bridge that acts as a crank neck.

The cranks are connected to each other using cylindrical elements that perform the functions of frame journals. At the same time, the geometry of the cheeks should ensure their reliable compression.

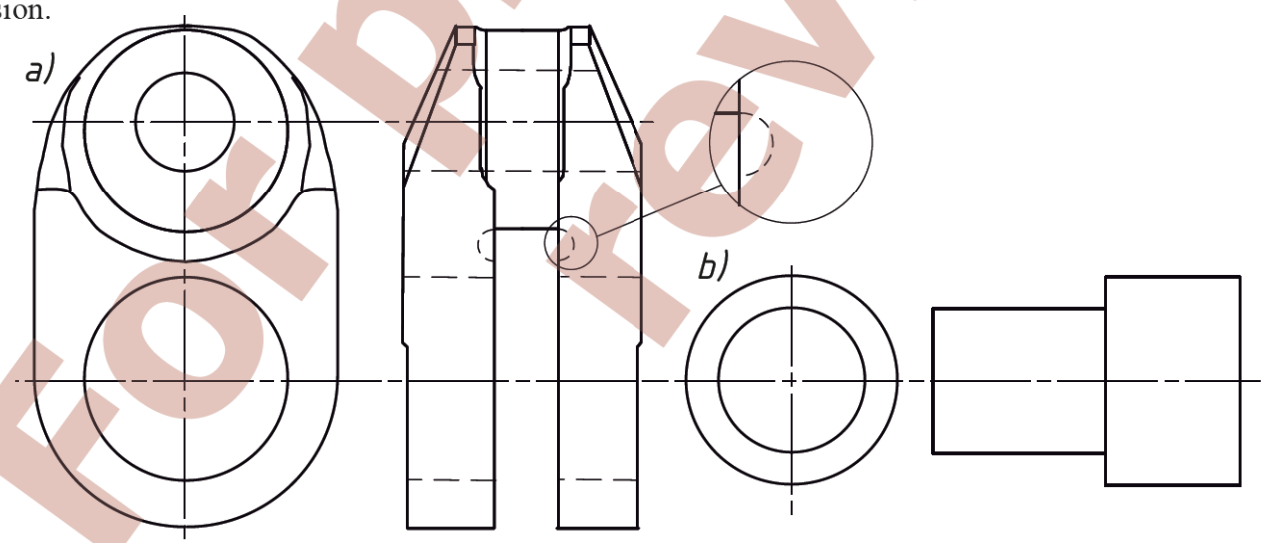


Figure 3.45 – Main elements of a semi-compound crankshaft of a low-speed diesel engine

Transitions between individual shaft elements are made smooth in the form of roundings called fillets (Fig. 3.44 *a*). In some designs, to reduce stress concentrations, the neck-cheek joint is made with two radii. To do this, the fillets are made recessed into the neck of the shaft (Fig. 3.44 *b*) or

For preliminary review

into the cheek (Fig. 3.45). In the latter case, it is possible to increase the supporting surface of the connecting rod journal.

Counterweights serve to unload the main bearings from the centrifugal forces of inertia and the moments they create resulting from the rotation of the unbalanced masses of the cranks. They are placed on the back of the cheeks.

Counterweights are most often attached to the cheeks with bolts or pins, to relieve which from shearing forces various types of locks, pins or keys are used. For small crankshafts, the counterweights are manufactured as one piece with the cheeks.

To reduce the mass of the counterweight, its center of gravity is located as far as possible from the shaft axis. In HSEs, the counterweights of the outer crank sometimes serve as a torsional vibration damper.

At the rear end of the crankshaft there is a flywheel mounting flange, which is made integral with the shaft or separately pressed onto it (Fig. 3.46).

The free end of the shaft is usually used for mounting the drive gear of pumps mounted on diesel engines (oil, water, fuel priming) and other auxiliary mechanisms.

To prevent oil leaks, the crankshaft exits from the crankcase cavity are sealed (Fig. 3.46). For this purpose, rubber-metal oil seals, asbestos fiber packing, oil scraper rings, oil drain threads, etc. are used. To reduce the ingress of oil onto the sealing elements, oil baffle washers are installed in front of them on the crankcase side.

For the manufacture of crankshafts, the highest quality alloy steels are used, less often modified cast irons with nodular graphite are used.

To harden the working surfaces of crankshaft elements, hardening of the journals and shaft fillets with high-frequency currents followed by low-temperature tempering is widely used. A significant increase in the service life of crankshafts is achieved through nitriding, during which the fatigue strength increases by 1.5...2 times, and the wear resistance of the journals increases by more than 20%.

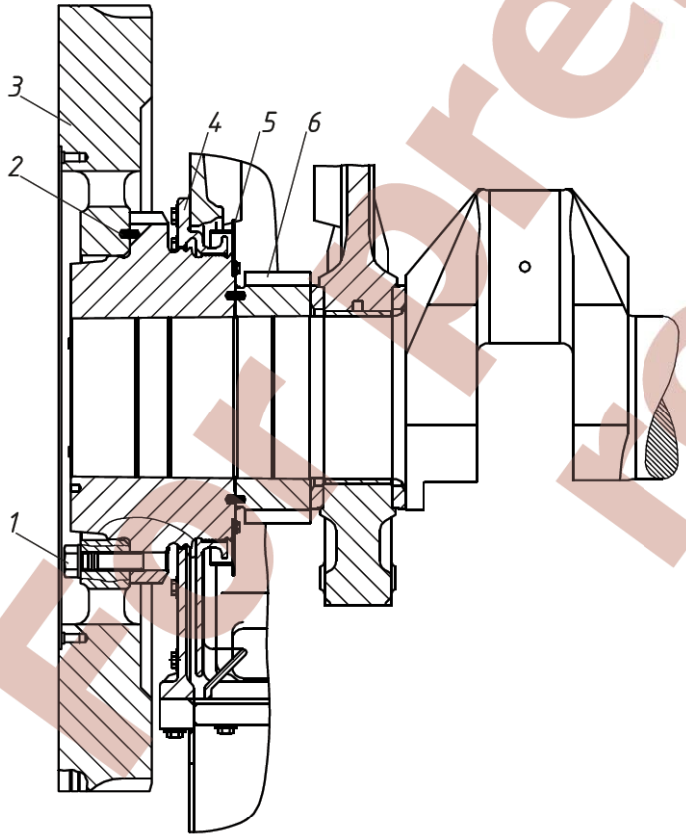


Figure 3.46 – Attaching the flywheel to the engine crankshaft: 1 – flywheel mounting bolts; 2 – installation pin; 3 – flywheel; 4 – shaft seal flange; 5 – oil squeegee; 6 – auxiliary mechanism drive gear (adapted from [45])

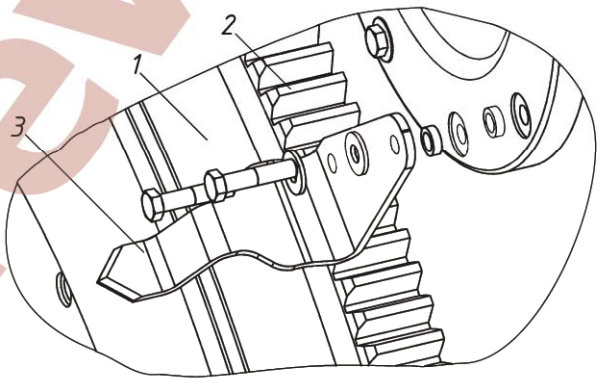


Figure 3.47 – Flywheel scale marker: 1 – flywheel; 2 – ring gear; 3 – scale marker (adapted from [45])

The flywheel serves to smooth out uneven torque and is a flat disk of large mass. During the working stroke of the piston, the flywheel accumulates mechanical energy, and during non-working strokes it releases it, thereby reducing fluctuations in the angular velocity of the crankshaft. In

Preliminary review

multi-cylinder engines, auxiliary strokes are made due to working strokes in adjacent cylinders, so there is no need for a massive flywheel in such engines. In this case, only the gear for the turning device is mounted on the crankshaft.

The flywheel is usually attached to the rear flange of the crankshaft using bolts. The position of the flywheel is fixed by locating pins located asymmetrically to eliminate the possibility of incorrect installation. In addition, the pins relieve the flywheel mounting bolts from shearing forces.

A ring gear is located on the flywheel rim, which meshes with a shaft turning device used to rotate the engine shaft during maintenance and repair. For low-power HSEs, the flywheel crown is used during start-up to crank the engine using an electric or pneumatic starter.

The TDC marks of all cylinders and an angular scale for checking the valve timing and fuel supply are applied to the cylindrical surface of the flywheel (Fig. 3.47). In diesel generators, the flywheel is used to connect the engine to the generator coupling.

During operation, the crankshaft perceives forces of variable magnitude and direction resulting from gas pressure on the piston, inertial forces of reciprocating and rotationally moving masses, and moments created by these forces. Under the influence of all these factors, the shaft is subject to bending and torsion deformations. Periodic sign-alternating relative angular displacements of the crankshaft elements, occurring under the influence of time-variable and phase-shifted torques on individual cranks, are called *torsional*. The greatest danger to the crankshaft is resonant vibrations, that is, when forced vibrations of the shaft coincide with its own vibrations. The crankshaft rotation speeds at which resonance occurs are called *critical.*

In addition to torsional vibrations, under the influence of disturbing factors the shaft undergoes axial vibrations, which can disrupt the normal operation of the auxiliary mechanism drives.

To combat various types of vibrations of the crankshaft, special devices called dampers or dampers are installed on it. They eliminate these fluctuations or significantly reduce their amplitude.

3.3 Crankshaft bearings

3.3.1 Frame (main) bearings of the crankshaft

Crankshaft bearings are the most important engine components that influence its reliable operation and service life. According to their purpose, frame bearings are divided into support and thrust bearings. Support bearings limit the radial movements of the crankshaft, and thrust bearings limit the axial movements.

As support bearings for modern marine diesel engines, plain bearings are used, which are made in the form of steel liners filled with an antifriction layer of white metal or a tin-aluminium alloy.

In modern LSEs, the liners are installed in pairs, in the bore of the foundation frame and the cover (Fig. 3.4 *a*), or only in the bore on the engine foundation frame (Fig. 3.4 *b*).

Depending on the thickness of the steel base, bearings are divided into thick-walled and thin-walled.

Thick-walled liners have a steel base of sufficient rigidity (Fig. 3.48) to ensure stability of the geometric shape and maintain an anti-friction layer in areas where the liner does not have support, for example, in the area of the mating surfaces of the upper liner.

In some designs, for precise positioning of the liners relative to each other, during assembly they are tightened with tight-fitting bolts. To adjust the vertical gaps between the joints of the liners, spacers are installed.

Thin-walled bearing shells have a thickness of 2...2.5% of the journal diameter (Fig. 3.49). They do not have sufficient rigidity, so they must rest tightly along their entire length. For this purpose, bearings of this type are manufactured with a slightly increased circumference, due to which, when installing the liners and tightening the bearing cap, the necessary radial pressure is created between the liner and the bearing housing. The liners are secured against rotation by special cylindrical pins installed in a bore that overlaps the bearing housing and the liner.

Regardless of the design, an oil distribution groove is made on the inner surface of the upper

For preliminary review

liner, into which oil is supplied under pressure. The lower liner has a tangential outlet for the oil distribution groove on the surface of the liner (section *A-A*, Fig. 3.48, 3.49). The presence of a smooth exit prevents the removal of oil from the shaft by the edge of the lower liner and reduces the resistance to oil flow when passing into the loaded area of the bearing.

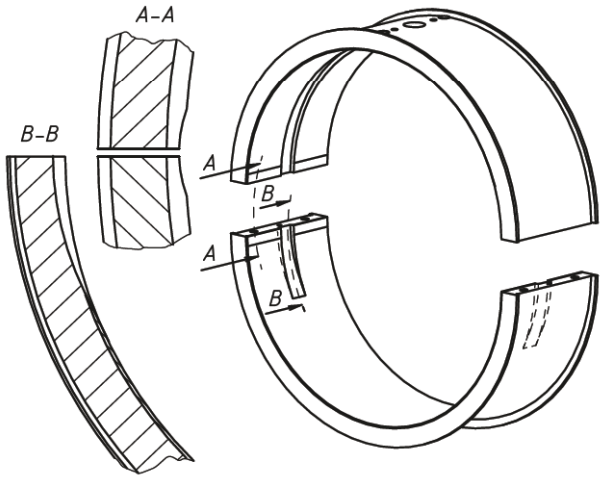


Figure 3.48 – Thick-walled liner for engines of the K, L, S series – MC from MAN (adapted from [19])

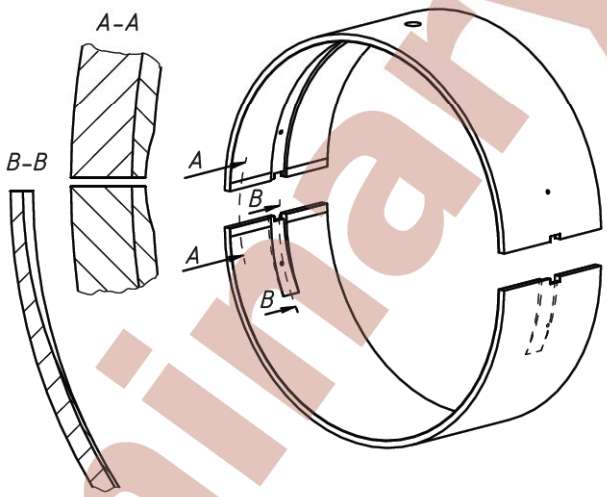


Figure 3.49 – Thin-walled liner for engines of the K, L, S series – MC-C from MAN (adapted from [19])

Near the joining surfaces of the upper and lower liners, thinnings (pockets) are made (section *B-B*, Fig. 3.48, 3.49). Their purpose is to compensate for inaccurate joining, which may result in the formation of a protrusion (step) on the joining surface. This protruding edge can remove oil and impair lubrication conditions in the bearing. In addition, the presence of pockets increases oil flow through the bearing, which promotes better removal of heat and wear products from the friction zone.

The main bearings of four-stroke MSE and HSE are made in the form of two thin-walled liners, the design of which is similar to the LSE bearings.

Main bearing shells are usually three-layer. The working layer is an alloy of tin *Sn* and antimony *Sb*, which has high anti-friction properties. To speed up the running-in of the bearing, as well as to protect it from corrosion, a thin (~3 μm) layer of tin is applied to the working surface. A layer of copper or nickel is used as a substrate to ensure better adhesion of the working layer to the steel base.

Recently, on highly accelerated engines, main bearings with groove-type liners have become widespread (Fig. 3.50).

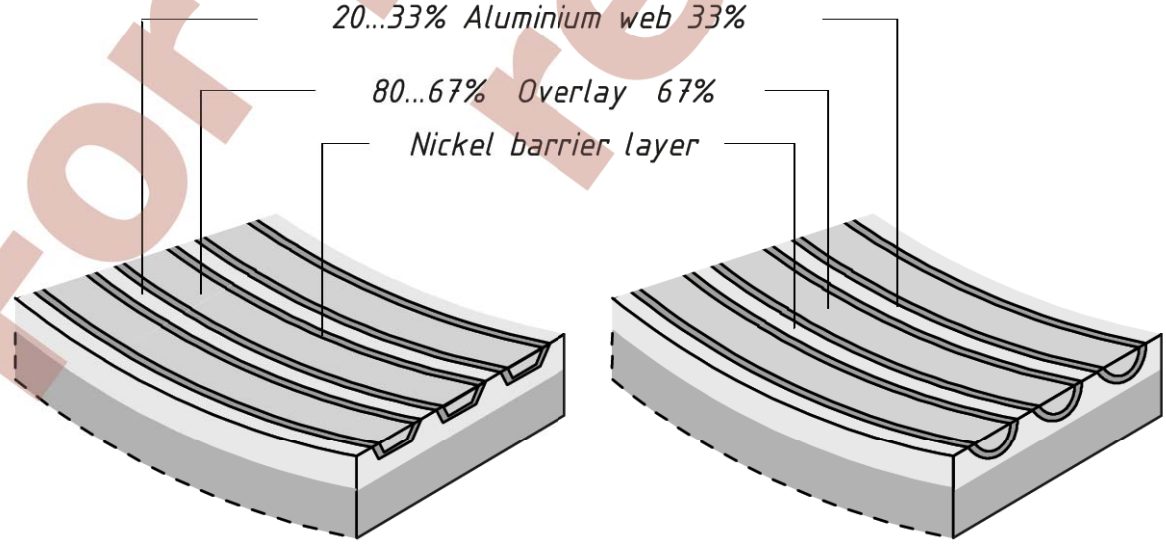


Figure 3.50 – Groove type main bearings (adapted from [46])

For preliminary review

Unlike multilayer bearings with a continuous working surface, in bearings of this type the working surface is formed from alternating grooves and projections. In this case, the grooves are filled with a soft tin-lead alloy, and the protrusions are made of a harder aluminium alloy with high load-bearing capacity and wear resistance. For better adhesion of the layers to each other, the grooves are coated with a thin layer of nickel. In this case, about 67...80% of the area of the working surface of the liner is occupied by grooves; about 20...33% by aluminium ribs and a maximum of 5% by nickel bridges between them.

This design reduces the likelihood of scoring, due to the fact that solid inclusions, entering the friction zone, are pressed into the soft surface and lose mobility, and the presence of nickel partitions prevents chipping of large areas of the working surface. As a result, the service life of such bearings is 1.5...2.5 times higher than that of traditional three-layer bearings.

3.3.2 Crankshaft thrust bearings

Thrust bearings serve to secure the crankshaft from axial movement.

For four-stroke MSE and HSE operating on a gearbox or generator, where there are no high axial loads, semi-ring washers are used as thrust bearings, installed on one of the main supports of the engine crankcase. The half rings are made of steel with an anti-friction coating on the working surface. They are secured from turning with special stoppers (Fig. 3.51 *a*) or pins (Fig. 3.51 *b*).

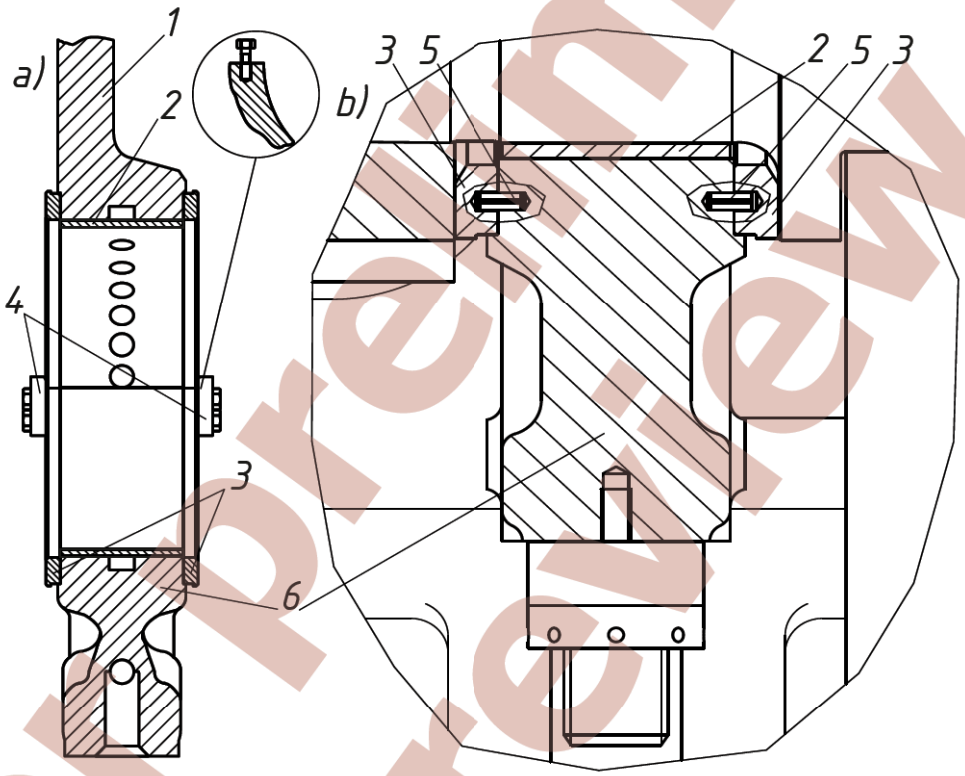


Figure 3.51 – Fixation of thrust bearings against rotation: *a* – engine 46 series from Wärtsilä; *b* – MaK 43; 1 – bearing housing; 2 – radial bearing shell; 3 – thrust bearing shells; 4 – fixing bar; 5 – fixing pin; 6 – bearing cap (adapted from [1, 12, 45])

The LSE thrust bearing usually performs the functions of the main thrust bearing, that is, it receives axial force from the screw stop. Therefore, more stringent requirements are placed on its design.

In modern LSEs, the thrust bearing is built into the rear end of the engine and is usually combined with a chain or gear drive of auxiliary mechanisms. Floating segments of the Mitchell type are used as thrust surfaces (Fig. 3.52).

On both sides of the thrust ridge of the crankshaft, several segments are installed, which are kept from turning by locking caps. During operation, the segments have a certain degree of freedom, thanks to which each of them can independently occupy the entry angle necessary to form a hydro-

For preliminary review

dynamic oil wedge. Oil for lubrication and cooling is supplied directly to the front and rear sides of the thrust comb through nozzles located in the spaces between the pads. The nozzles are mounted on a semi-circular supply pipe.

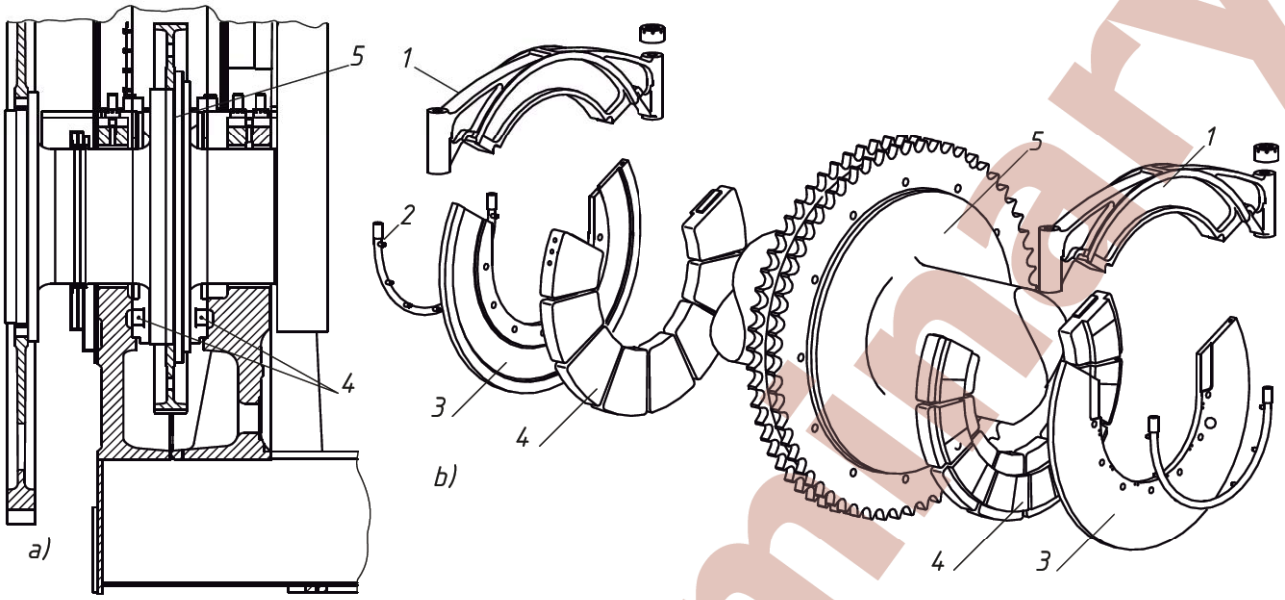


Figure 3.52 – Thrust bearing assembly for a low-speed diesel engine of the X72 series from WinGD (*a*) and the K, L, S series from MAN (*b*); 1 – top cover; 2 – lubricating oil supply tube; 3 – thrust plate; 4 – thrust segments; 5 – thrust comb (adapted from [21])

3.4 Kinematics, dynamics of the crank mechanism and balancing of marine engines

3.4.1 Kinematics of the crank mechanism

The crank mechanism is used to convert the reciprocating motion of the piston into the rotational motion of the crankshaft. Below are the kinematic dependencies for the crankshafts that are most widely used in marine internal combustion engines.

In Fig. 3.1 shows diagrams of crank mechanisms most often used in marine engines.

Most designs use a central crankshaft (Fig. 3.53 *a*), in which the axes of the cylinder and crankshaft are in the same plane.

Such a mechanism is characterized by two kinematic parameters: the crank radius r and the connecting rod length L_{cr}. Instead of these parameters, a dimensionless quantity is usually used – the mechanism constant λ_{cr} (clause 2.2):

$$\lambda_{cr} = r/L_{cr}. \quad (3.1)$$

At large values of λ_{cr} (relatively short connecting rods), the height and weight of the engine decrease, but the normal pressure of the piston on the cylinder liner (or the crosshead slider on the guides) and the second-order inertia forces of the translational-moving masses increase.

An offset (disaxial) crankshaft (Fig. 3.53 *b*) can be obtained by shifting the pin axis relative to the cylinder axis or by shifting the cylinder axis relative to the crankshaft axis. Displacement (displacement) is indicated by the symbol e.

Small desaxing ($e < 0.03r$) is provided to reduce the noise that occurs when moving the piston in the cylinder, and is realized by shifting the axis of the bore of the hole for the piston pin relative to the piston axis. In this case, it is permissible to perform kinematic calculations using the formulas for central crankshafts.

Deoxidation within the limits of $e = (0.1...0.4)r$ is used to reduce the difference in piston pressure on the right and left sides of the cylinder, which gives more uniform engine wear. In addition, the disaxial mechanism is characterized by a piston speed near TDC that is lower than that of the central crankshaft, which improves the combustion process. At the same time, the inertia forces of the reciprocating moving masses of the disaxial crank mechanism are slightly higher than the central one. Minor deoxygenation is present in most engines with a V-shaped cylinder arrangement. True,

For preliminary review

the influence of this deoxygenation is insignificant and in practical calculations formulas for the central crank mechanism are usually used.

Nowadays, disabling in excess of $e = 0.1r$ is used only in engines with special kinematic designs (for example, in engines with a rhombic mechanism or in engines with a parallel arrangement of each pair of cylinders operating on one crank.

Marine V-shaped engines use mechanisms with trailed connecting rods. A schematic diagram of a crank mechanism with a trailed connecting rod is shown in Fig. 3.53 c. One connecting rod in this mechanism is pivotally connected directly to the crankshaft journal and is called the main connecting rod, and the second is connected to the main connecting rod by means of a pin located on its head and is called the trailing rod (Fig. 3.35 *g*).

The system of the main and trailing connecting rods is used, for example, on 7FDS diesel engines from General Electric. In such a connecting rod system, high rigidity of the crank head of the main connecting rod is noted; however, the pistons connected to the main and trailing connecting rods have unequal strokes, since the axis of the crank head of the trailing connecting rod during operation describes an ellipse, the major semi-axis of which is greater than the radius of the crank (Fig. 3.53 *c*). The kinematics and dynamics of these mechanisms are covered in sufficient detail in the specialized literature; consideration of this issue is beyond the scope of this book.

When calculating for a central type crank mechanism (Fig. 3.53, *a*), the relationship between the piston movement S_x and the crankshaft rotation angle φ is determined as follows:

$$S_x = A'A = A'O - AO = (A'B' + B'O) - (AC + CO). \tag{3.2}$$

The segment $A'B'$ is equal to the connecting rod length L_{cr}, and the segment $B'O$ is equal to the crank radius r. Taking this into account, as well as expressing the segments AC and CO through the product L_{cr} and r, respectively, by the cosines of angles β and φ, we obtain

$$S_x = (L_{cr} + r) - (L_{cr}\cos\beta + r\cos\varphi) = r\left[1 - \cos\varphi + \frac{\lambda_{cr}}{4}(1 - \cos\beta)\right]. \tag{3.3}$$

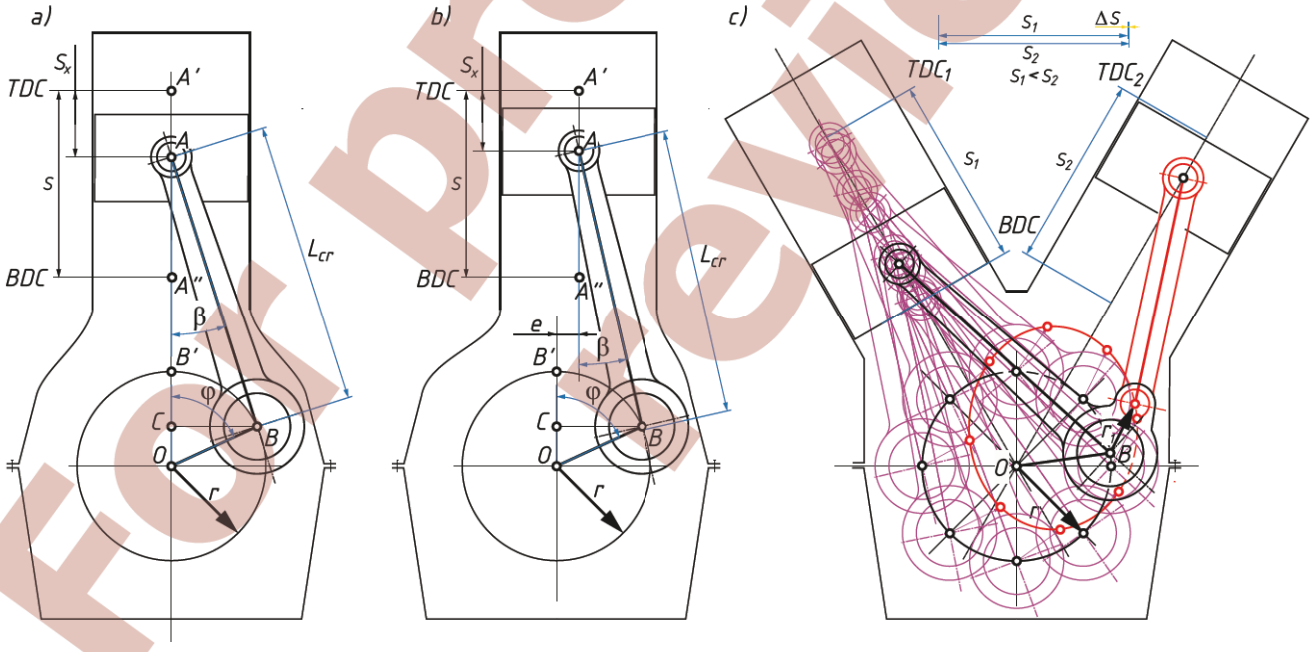


Figure 3.53 – Diagrams of crank mechanisms of internal combustion engines; *a* – central; *b* – displaced (disaxial); *c* – with trailed connecting rod

From the triangles ACB and OCB we find $CB = AB \sin\beta = OB \sin\varphi$ or $L_{cr}\sin\beta = r\sin\varphi$, from which:

For preliminary review

$$\sin\beta = \frac{r}{L_{cr}}\sin\varphi = \lambda_{cr}\sin\varphi. \quad (3.4)$$

Expression (3.4) includes cos β. Let's express this function through sine:

$$\cos\beta = \sqrt{1-\sin^2\varphi} = \sqrt{1-\lambda_{cr}^2\sin^2\varphi}. \quad (3.5)$$

Let us expand expression (3.5) into a series using Newton's binomial, and we obtain:

$$\cos\beta = 1 - \frac{\lambda_{cr}^2}{2}\sin^2\varphi - \frac{\lambda_{cr}^4}{8}\sin^4\varphi - \frac{\lambda_{cr}^6}{16}\sin^6\varphi - \ldots \quad (3.6)$$

For values of λ_{cr} characteristic of marine engines in the expansion formula, the value of terms including this value to a power higher than the first quickly decreases. For example, with $\lambda_{cr} = 0.25$ and $\varphi = 90°$, the second term of the expansion is about 3%, and the third term is only about 0.05% of the value of the first term of the expansion. This allows us, with sufficient accuracy for practice, to limit ourselves to the first two terms of the series (3.6), i.e., the cosβ function is expressed through sin φ as follows:

$$\cos\beta = 1 - \frac{\lambda_{cr}^2}{2}\sin^2\varphi. \quad (3.7)$$

Considering that

$$\sin^2\varphi = \frac{1}{2}(1-\cos 2\varphi),$$

expression (3.7) can be written as:

$$\cos\beta = 1 - \frac{\lambda_{cr}^2}{2}(1-\cos 2\varphi). \quad (3.8)$$

Substituting (3.8) into equation (3.3), we obtain an approximate expression for determining the piston stroke:

$$S_x = r\left[(1-\cos\varphi) + \frac{\lambda_{cr}}{4}(1-\cos 2\varphi)\right]. \quad (3.9)$$

Analysis of the above formulas shows that the path of the piston S can be conditionally represented as consisting of two harmonic movements (harmonics of the 1st and 2nd order) (Fig. 3.54):

$$S = S_1 + S_2.$$

Movement of the piston due to the rotational movement of the crank (1st order harmonic).

$S_1 = r\ (1 - \cos\varphi)$ – piston movement with an infinitely long connecting rod ($L_{cr} = \infty$), when $\lambda_{cr} = 0$;

Movement of the piston due to the rocking motion of the connecting rod (second order harmonic).

$S_2 = \lambda_{cr} r\ /4(1 - \cos 2\varphi)$ – correction for piston movement taking into account the final length of the connecting rod.

So, for example, at $\lambda_{cr} = 0.25$, turning the crankshaft at an angle $\varphi = 90°$ (the first quarter of the circle) leads to the fact that the piston travels more than half of its stroke (0.56 S).

This is explained by the fact that when φ changes from 0 to 90°, the connecting rod moves towards the crankshaft and at the same time deviates from the cylinder axis, both of these movements corresponding to the movement of the piston in the same direction.

When φ changes from 90 to 180°, the connecting rod no longer moves away, but approaches the cylinder axis, which counteracts the movement of the piston. Thus, when the angle of rotation of the

For preliminary review

crankshaft changes in this section, the piston travels less than half of its stroke (0.44 *S*).

We obtain an approximate formula for determining the piston speed by differentiating expression (3.9) with respect to time:

$$W = \frac{dS_x}{dt} = \frac{dS_x}{d\varphi}\frac{d\varphi}{dt} = r\frac{d\varphi}{dt}\left(\sin\varphi + \frac{\lambda_{cr}}{2}\sin 2\varphi\right). \qquad (3.10)$$

In the kinematic analysis of the crank mechanism, it is assumed that the rotation speed of the crankshaft is constant. In this case:

$$\omega = \frac{d\varphi}{dt} = \frac{\pi n}{30} \qquad (3.11)$$

where ω – angular velocity of the crankshaft, rad/s; *n* – crankshaft rotation speed, min^{-1}.

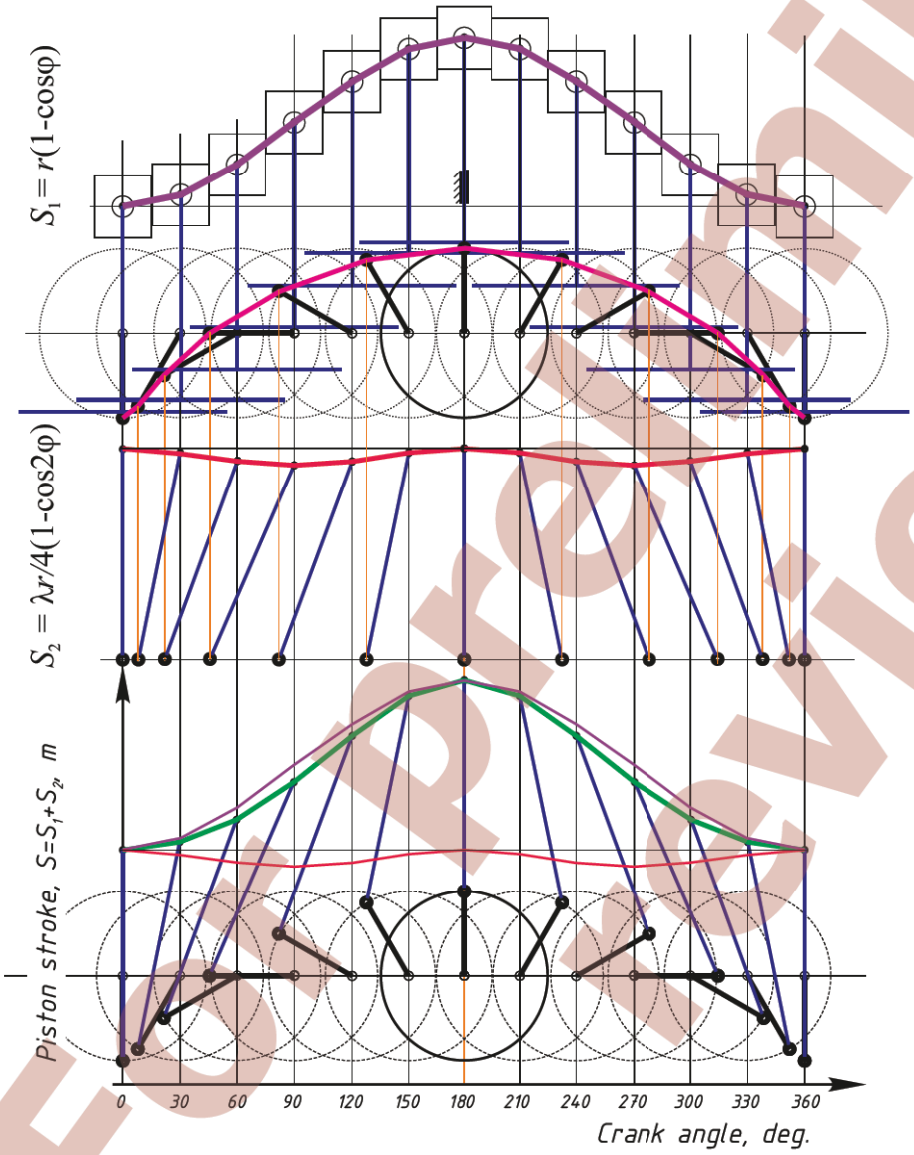


Figure 3.54 – Piston movement as the sum of movements of the rotational movement of the crank and the rocking movement of the connecting rod

Taking (3.11) into account, equation (3.10) takes the form:

$$W = r\omega\left(\sin\varphi + \frac{\lambda_{cr}}{2}\sin 2\varphi\right). \qquad (3.12)$$

The piston speed, as well as the piston path, can be represented as the sum of two terms (Fig. 3.55):

$$W = W_1 + W_2.$$

For preliminary review

Movement of the piston due to the rotational movement of the crank.

$W_1 = rW\sin\varphi$ – harmonically varying speed of the piston of the first order, that is, the speed with which the piston would move if there was a connecting rod of infinitely long length ($L_{cr} = \infty$; $\lambda_{cr} = 0$);

$W_2 = ((\lambda_{cr} rW)/4)\sin 2\varphi$ – harmonically varying speed of the piston of the second order, that is, the speed from the additional movement of the piston, due to the finite length of the connecting rod.

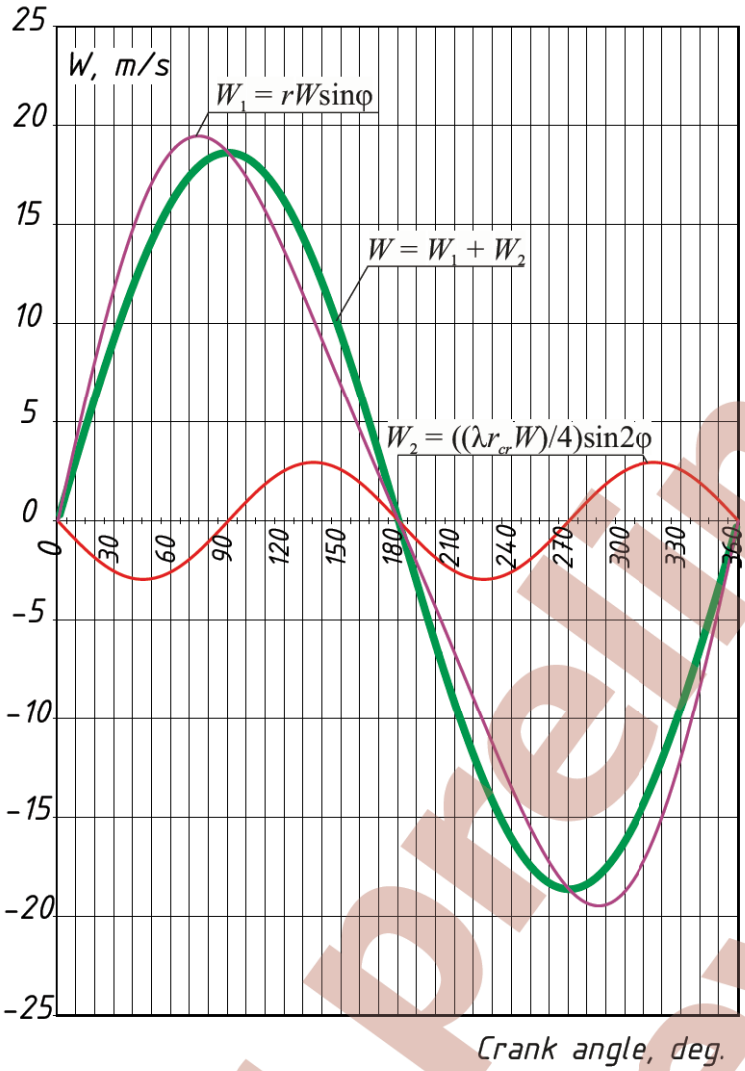


Figure 3.55 – Change in piston speed W depending on the crankshaft rotation angle φ for a high-speed diesel engine

The highest piston speed c_{max} is greater than the maximum peripheral speed of the center of the connecting rod journal W_{max} by 2...5% (depending on the value of λ_{cr}). In practice, when assessing mechanical losses in an engine, they use such an indicator as the average piston speed c_p, which can be found from the expression:

$$c_p = \frac{2Sn}{60} = \frac{Sn}{30} = \frac{rn}{15}$$

For most modern low-speed marine engines c_p = 7.5...8.8 m/s for medium- and high-speed ones c_p = 8.0...10.2 m/s.

Having differentiated (3.12) with respect to time, we obtain an expression for determining the acceleration of the piston (Fig. 3.56):

$$J = \frac{dW}{dt} = r\omega^2 \cos\varphi + \lambda_{cr} r\omega^2 \cos 2\varphi = r\omega^2 (\cos\varphi + \lambda_{cr} \cos 2\varphi). \qquad (3.13)$$

Where j_1 = cos φ – first order harmonic component; $j_2 = \lambda_{cr}\, r\omega^2 \cos 2\varphi$ – second-order harmonic component.

For preliminary review

3.4.2 Dynamics crank mechanism

When the engine is running, the following forces act on the crank of each individual cylinder:

- action of gases on the piston P_g;
- inertia of moving parts P_j;
- severity P_{gr};
- friction P_f;
- air resistance from the crankcase P_{atm}.

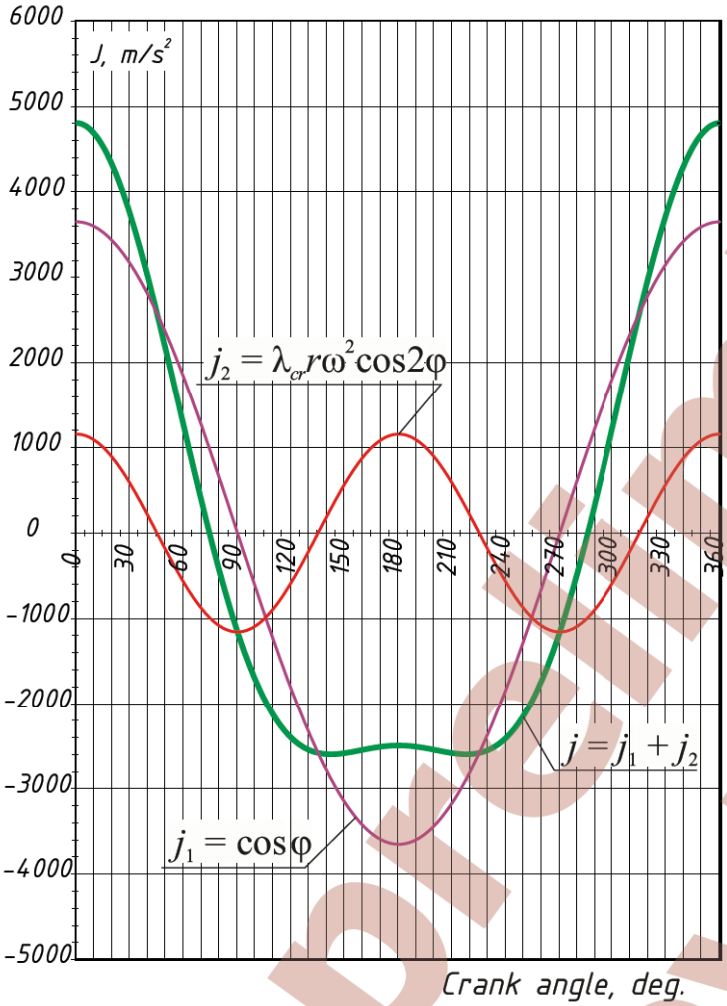


Figure 3.56 – Change in acceleration J of the piston depending on the angle φ of rotation of the crankshaft for a high-speed diesel engine

The forces P_{gr}, P_f, P_{atm} are relatively small, so they are usually neglected.

Forces from gas pressure P_g are associated with changes in pressure in the working cylinder, inertia forces P_j are determined by the masses of the moving parts of the crank mechanism, the piston stroke and the diesel engine speed.

Finding these forces is necessary to calculate the strength of diesel parts, assess the loads on bearings, determine the degree of uneven rotation of the engine crankshaft and calculate it for torsional vibrations.

When constructing a diagram of the forces acting in the crank mechanism, the initial force is the specific total force P_c acting on the pin (crosshead cross member). It represents the algebraic sum of the gas pressure forces P_g acting on the piston bottom, and the specific inertia forces P_j of the masses of parts moving back and forth,

$$P_s = P_g + P_j. \tag{3.14}$$

Let us consider in more detail the action of gas pressure forces on the piston P_g of inertia forces of moving masses P_j.

The force values from the gas pressure in the cylinder P_g are determined from the expression:

$$P_g = (p_i - p_0)\, F_r, \tag{3.15}$$

For preliminary review

where p_i – current value of pressure in the working cylinder (determined from the expanded $p\varphi$-diagram, MPa;

F_p – piston area, mm^2;

p_0 – pressure in the sub-piston cavity. For two-stroke crosshead engines $p_0 = p_s$, for four-stroke trunk engines p_0 is taken equal to atmospheric pressure (0.1 MPa).

Due to the peculiarities of the kinematic diagram, the crank mechanism has two conventional systems of masses – translationally moving masses – m_j, which include the piston, piston pin, part of the mass of the connecting rod, and unbalanced rotating masses – m_R.

Thus, the actual masses of the moving parts are replaced by reduced masses, conditionally concentrated at those points of the mechanism whose laws of motion are known. In this case, the masses concentrated at the same point are summed up, and their action should be equivalent to the action of real masses (Fig. 3.57).

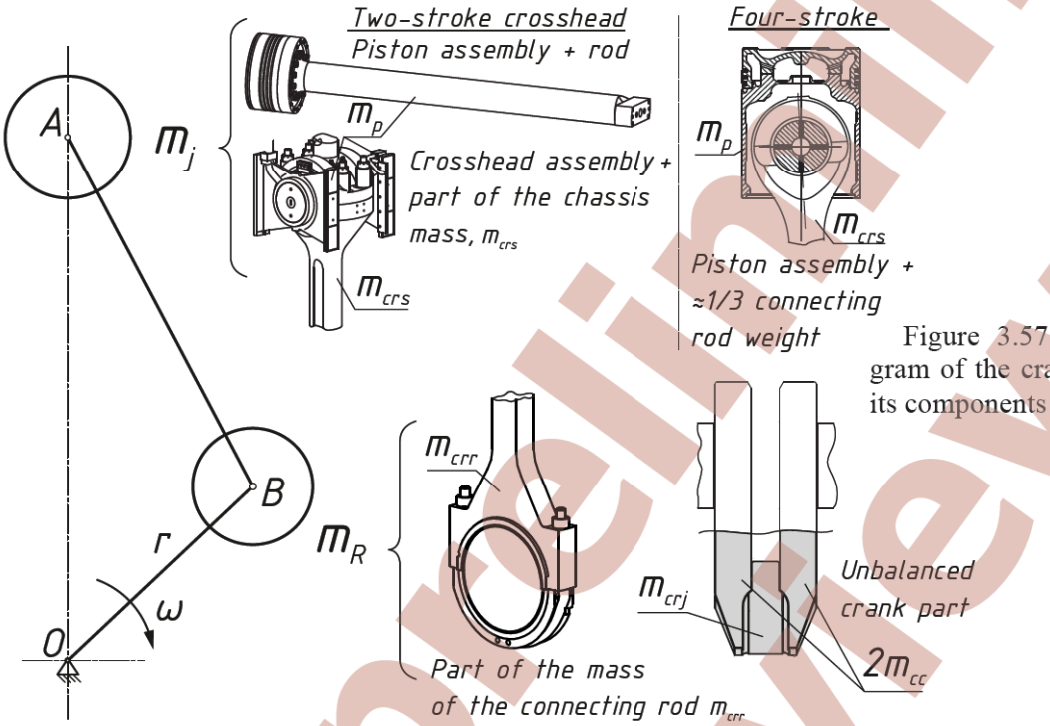


Figure 3.57 – Two-mass diagram of the crank mechanism and its components

The relative masses for points A and B (Fig. 3.57) are determined using the reduction scheme. Based on the fact that the connecting rod makes a complex movement in a vertical plane perpendicular to the axis of rotation, its mass m_{cr} is divided into two reduced masses m_{crs} and m_{crr}. The mass m_{crs} is concentrated in the center of the upper head of the connecting rod and performs translational motion, and the mass m_{crr} is located in the center of the crankpin of the crankshaft and performs rotational motion.

If we take m_{cr} as the mass of the connecting rod, and l_s as the coordinate of the center of its mass, the mass reduced to the piston pin axis will be determined by the formula:

$$m_{crs} = m_{cr}(l_s / L_{cr}) + m_p,$$

where m_p is the mass of the piston, which includes all the parts connected to it that perform translational motion along the axis of the working cylinder (Fig. 3.57). In crosshead diesel engines, the progressively moving masses also include the weight of the rod and the crosshead.

The unbalanced mass of the crank consists of the mass of two cheeks $2m_{cc}$ and the mass of the connecting rod journal m_{crj}. The mass of the crank pin m_{crj} is located at a distance r from the center of rotation and does not require reduction. Accordingly, m_{crr} and m_{cc} remain unbalanced. These unbalanced rotating masses, when reduced to the radius of the crank, are replaced by one m_r, defined by the formula:

For preliminary review

$$m_r = m_{cr}(L_{cr} - l_s)/L_{cr} + 2m_{cc}\, s'/r + m_{crj},$$

where s' is the coordinate of the center of mass of the crank cheek relative to the crank axis.

Depending on the design of the connecting rod, its reduced masses correspond to the following values: $m_{crs} = 0.3\ldots0.40\ m_{cr}$; $m_{crr} = 0.6\ldots07\ m_{cr}$; for crosshead LSEs $m_{crs} \approx m_{crr} \approx 0.5\ m_{cr}$.

The inertia force of the translationally moving masses P_j is applied at the center of the piston pin and acts along the cylinder axis, and the inertia force of the rotating masses P_R is applied at the center of the connecting rod journal and is directed along the crank from the center of rotation.

The specific inertial force of the crankshaft parts moving back and forth is determined from the relationship:

$$P_j = -m_j J; \qquad (3.16)$$

here J is the acceleration of reciprocating moving masses, m/s^2; m_j is the mass of parts performing reciprocating motion (Fig. 3.57).

In the TDC region, inertial forces impede the movement of the piston, and in the BDC region they promote it.

The driving or total force P acting in the crank mechanism and reduced to the axis of the piston pin (crosshead cross member) is the algebraic sum of the forces from the gas pressure P_g and the inertia of the translationally moving masses P_j, as well as the force of gravity of the translationally moving masses: $P_{gr} = m_j g$.

$$P = P_g + P_j + P_{gr}.$$

The curve of this force (Fig. 3.58), characterizing its change during the cycle, for a two-stroke engine over 360° and for a four-stroke over 720°.

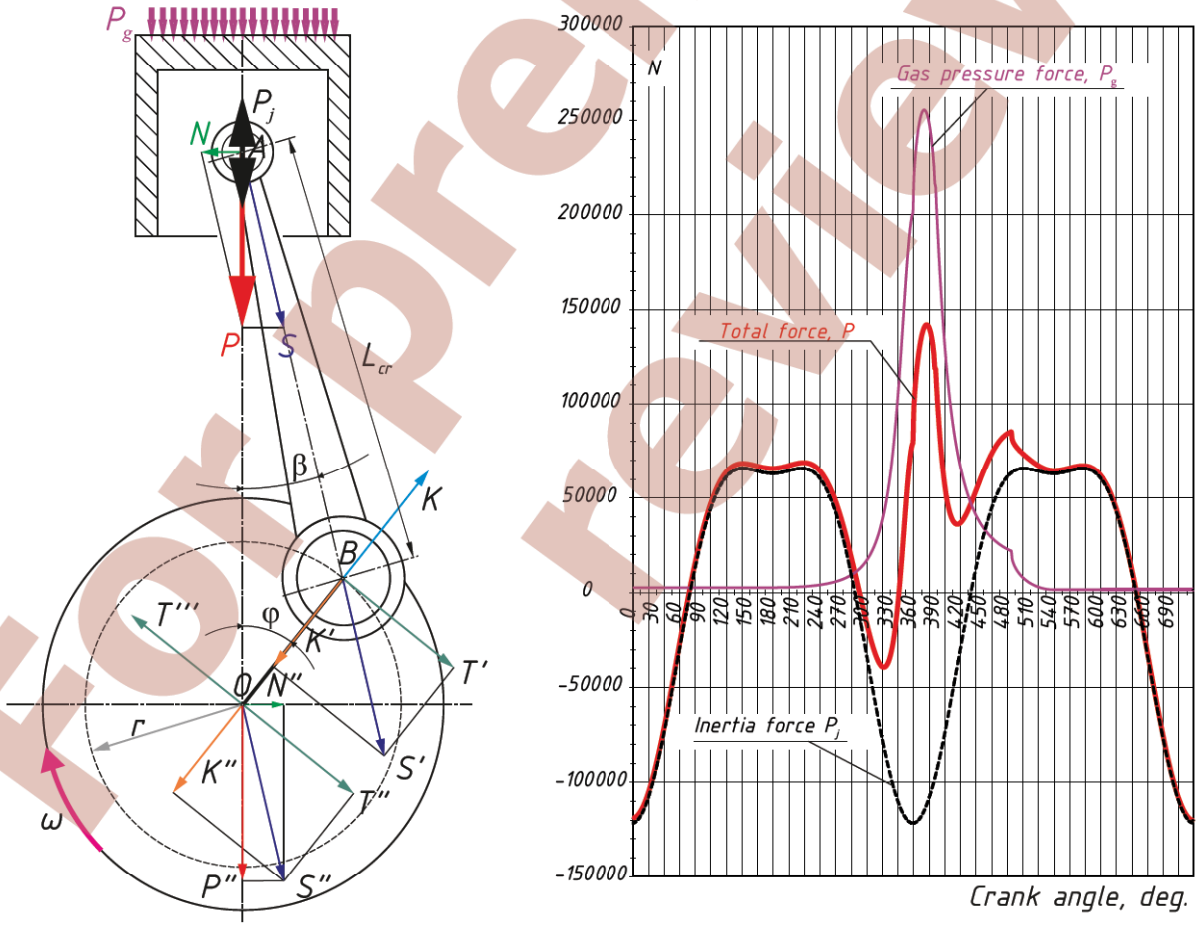


Figure 3.58 – Diagram of the decomposition of forces acting in the crank mechanism. The total force P reduced to the piston pin axis as the sum of the gas pressure forces P_g and the inertia of translationally moving masses P_j

For preliminary review

For four-stroke high- and medium-speed engines, the influence of the force P_{gr} is very insignificant, so it can be neglected. For low-speed engines, the mass of translationally moving masses can reach from several hundred kilograms to several tons, and therefore this force must be taken into account when assessing the total driving force of the piston.

Force P is decomposed into two components (Fig. 3.59):

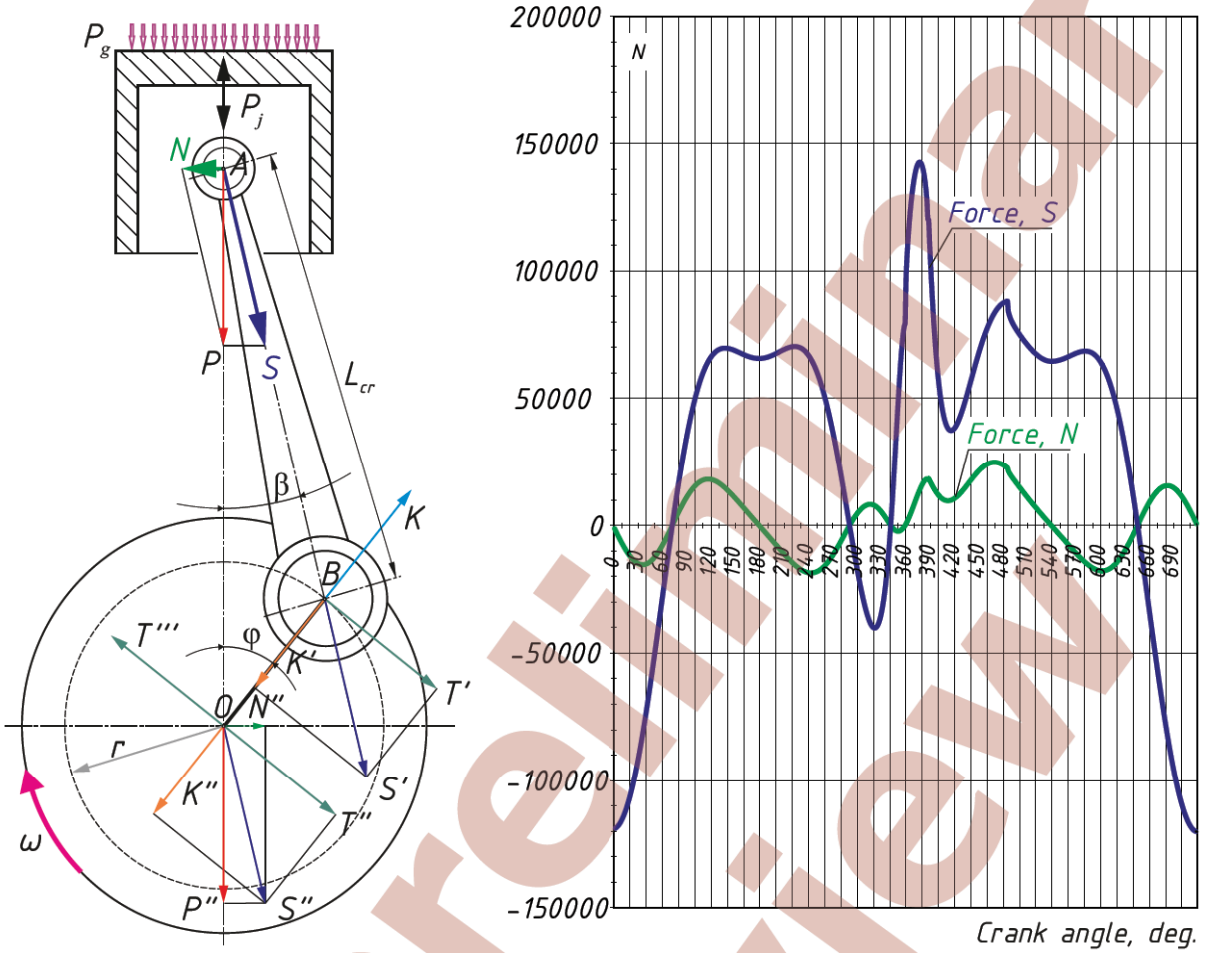


Figure 3.59 – Scheme of the decomposition of forces acting in the crank mechanism. Force S directed along the connecting rod and normal force N

- lateral force N perpendicular to the cylinder axis

$$N = P \operatorname{tg}\beta; \tag{3.17}$$

- force S directed along the axis of the connecting rod.

$$S = P / \cos\beta. \tag{3.18}$$

The connecting rod deflection angle β corresponding to the crank rotation angle φ can be found from the expression $\beta = \arcsin(\lambda_{cr} \sin \varphi)$.

Force N is also called *normal force*, it is variable in value and direction, and at TDC and BDC it is equal to zero. When the crank is turned from 0 to 180°, the force N is directed to the left, when the crank is turned from 180°to 360° – to the right. When the direction of the force changes, the piston (crosshead slider) shifts.

Power S, acting along the axis of the connecting rod, is always directed in one direction towards the axis of the connecting rod journal of the crankshaft and loads the bearings of the lower head of the connecting rod.

Let's transfer the force S along the line of its action to the center of the crank pin ($S' = S$) and decompose it according to the rule of vector decomposition into two components (Fig. 3.60):

- normal force K' directed along the radius of the crank,

For preliminary review

$$K' = S\cos(\varphi+\beta) = P\,(\cos(\varphi+\beta)/(\cos\beta)), \qquad (3.19)$$

- tangential force T' tangent to the radius of the crank:

$$T' = S\sin(\varphi+\beta) = P\,(\sin(\varphi+\beta)/(\cos\beta)). \qquad (3.20)$$

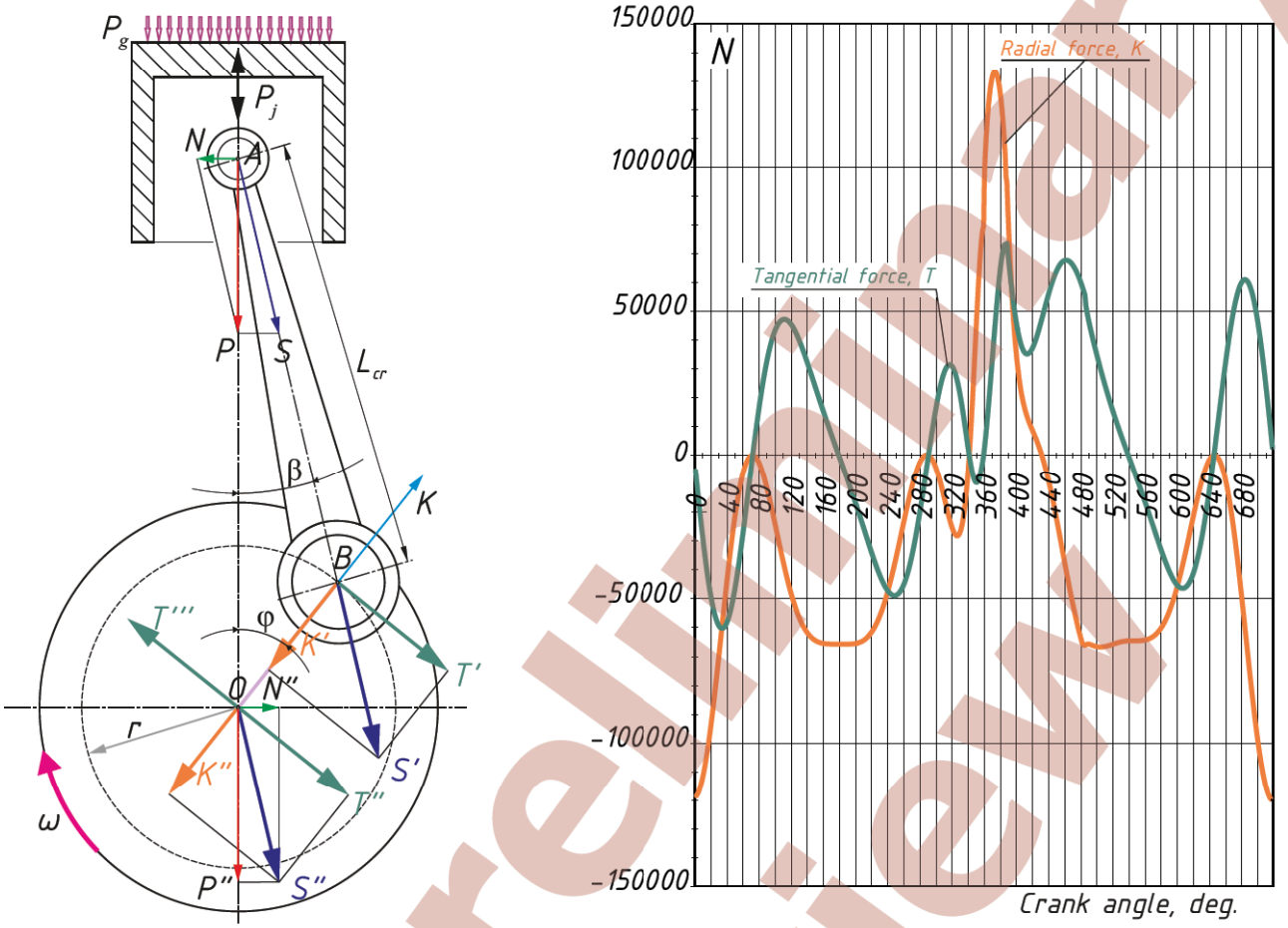


Figure 3.60 – Diagram of the decomposition of forces acting in the crank mechanism. Force decomposition S, directed along the connecting rod by force K directed along the radius of the crank and tangential force T

The normal force K' can be transferred along the line of its action to the center of the shaft and denoted as K'' ($K'' = K'$) (Fig. 3.60). To the center of the crank (point O) we apply two equal and oppositely directed forces T'' and T''' ($T' - T'' - T'''$) then a pair of forces T' and T''' on shoulder r will create moment called torque M_{tor} (Fig. 3.61 *a*). The forces T'' and T''' balance each other. The physical meaning of these two forces is that the force T'' is applied to the support, which is the main (frame) bearing of the crank, and the force T'' is the reaction of the support to this force, as a result of which the crank axis O remains motionless.

The torque is transmitted through the crankshaft to the flywheel and is equal to:

$$M_{tor} = TR = Pr\,(\sin(\varphi+\beta)/(\cos\beta)) \approx Pr\,(\sin\varphi - (\lambda_{cr}/2)\sin 2\varphi). \qquad (3.21)$$

Since the tangential force is variable (Fig. 3.60), the torque also changes from zero to maximum with a change in sign.

The forces K'' and T' can be added; their resultant S'', equal to the force S acting along the connecting rod, loads the main (frame) bearings of the shaft. The force S'' can also be decomposed into two components: N'', perpendicular to the cylinder axis, and P'', acting along the cylinder axis.

Forces N'' and N form a pair of forces (Fig. 3.61 *b*) on arm h, the moment of which is called overturning (M_{over}) and acts on the housing parts of the engine. The moment M_{over} is directed against the torque and, in accordance with the condition of equilibrium of the moving parts of the

For preliminary review

mechanism, is generally equal in value to the torque:

$$M_{over} = Nh = P \operatorname{tg}\beta\, r\, (\sin(\varphi+\beta)/(\sin\beta)) = Nr\, (\sin(\varphi+\beta)/(\cos\beta)). \quad (3.22)$$

The overturning torque tends to tilt the engine in the direction opposite to the action of the torque. Compensation for M_{over} is carried out by attaching the engine housing parts to a rigid support, massive enough to counteract this moment, for example, to the foundation of a ship, both directly and through elastic elements that serve to suppress vibrations arising during engine operation.

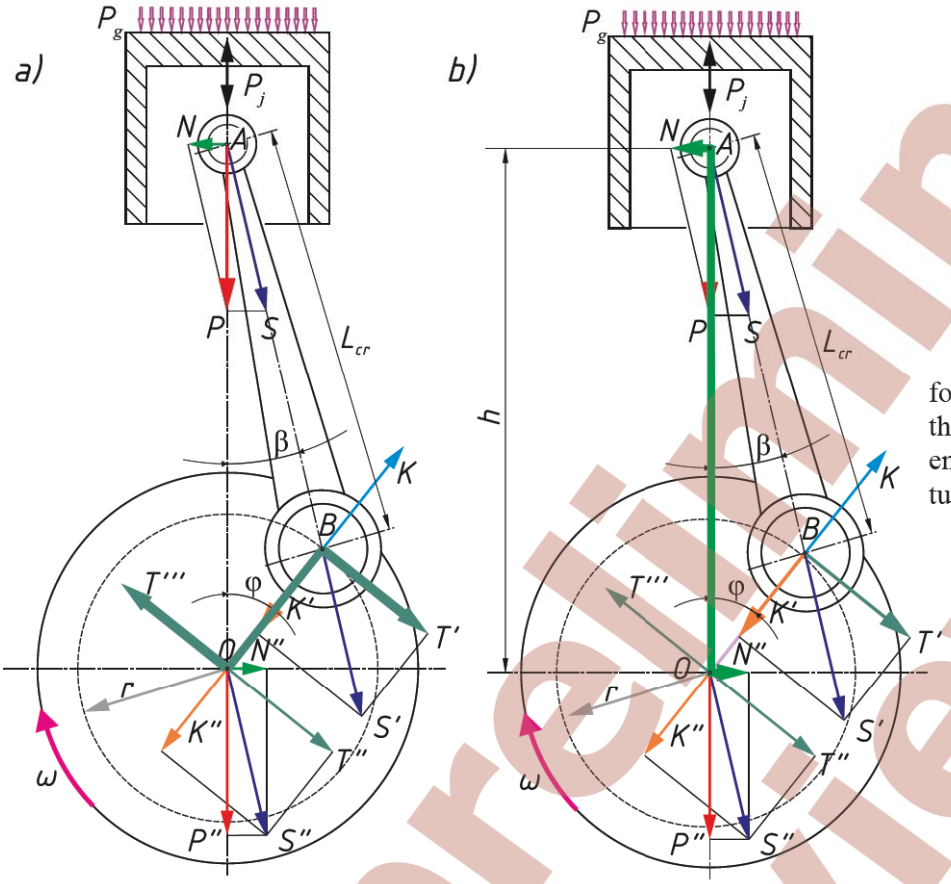


Figure 3.61 – Diagram of forces and moments acting in the crank mechanism of the engine: *a* – torque; *b* – overturning moment

In addition to the already discussed forces applied to the crank pin, it is also acted upon by the centrifugal force of inertia of the rotating masses, directed along the radius of the crank:

$$K = m_R r\, \omega^2. \quad (3.23)$$

At a steady engine speed, this force is constant in magnitude.

3.4.3 Total torque from all engine cylinders

In multi-cylinder engines, the crankshaft cranks perceive torques, which, summed up in accordance with the accepted operating order, give a total torque ΣM_{tor} at its end. The curve of this moment $\Sigma M_{tor} = f(\varphi)$ can be constructed by summing the torque curves M_{tor} for individual cylinders. In this case, the curves for individual cylinders must be shifted relative to each other by an angular interval θ, corresponding to the interval between working strokes in individual cylinders (crank wedge angle).

With equal intervals between flashes, the moment ΣM_{tor} has a period of change for four-stroke engines:

$$\theta = 720°/\, i,$$

for two-stroke:

$$\theta = 360°/\, i,$$

where i is the number of engine cylinders.

In Fig. 3.62 shows an example of finding ΣM_{tor} using the example of a four-cylinder, four-stroke engine with a crank angle of 180°.

For preliminary review

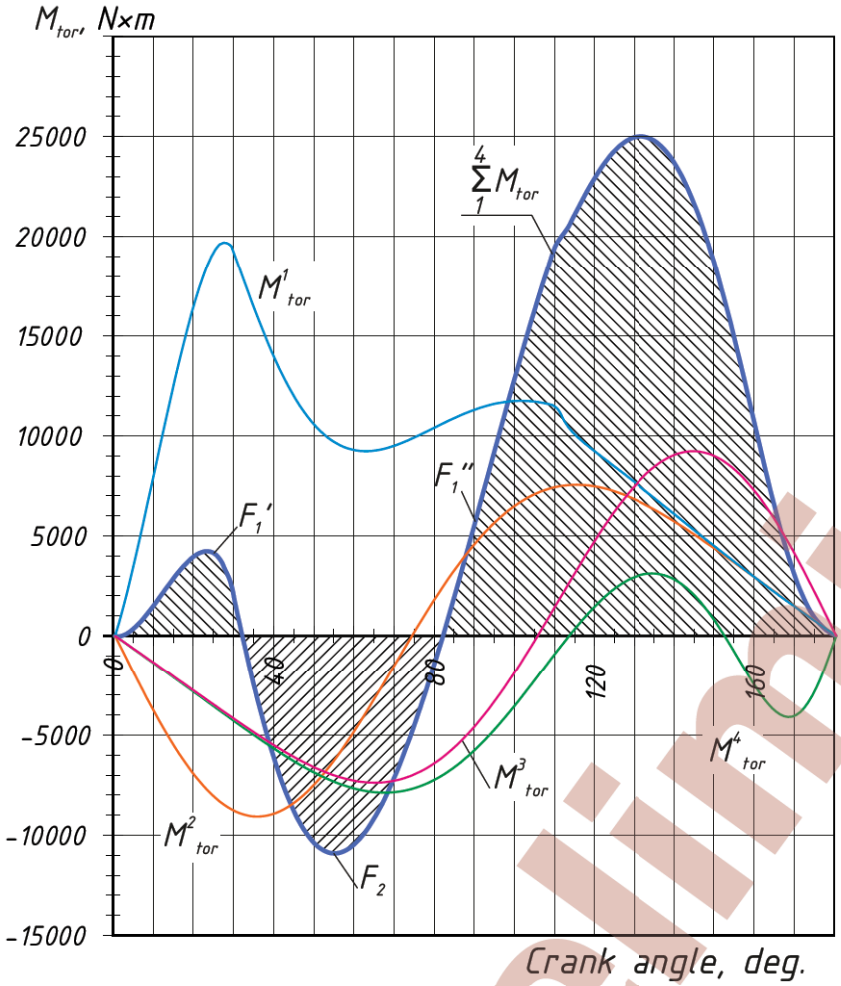


Figure 3.62 – Construction of the total torque ΣM_{tor} for a four-cylinder engine and determination of the average torque on the engine shaft M_{aver}

Average engine torque M_{aver}

The average value of the total engine torque is determined by the area limited by the torque curve, two – the ordinate and the abscissa,

$$M_{aver} = \frac{(F_2 - F_1)\mu_m}{l},$$

where F_1, F_2 are, respectively, positive and negative areas enclosed between the M_{aver} curve and the abscissa axis, equivalent to the work performed by the torque, mm^2;

μ_m – torque scale, (kN×m)/mm;

l – length of the diagram along the abscissa axis, mm.

In the example shown in Fig. 3.62, the area F_1 is the sum of the areas F'_1 and F''_1.

The average torque M_{ave} is an indicator, since when constructing a diagram of the torque M_z of an engine, friction and the costs of actuating auxiliary mechanisms and engine components are not taken into account. To obtain the effective torque M_e, kN×m, it is necessary to multiply it by the mechanical efficiency η_m, obtained by calculation or experiment:

$$M_e = M_{ave}\,\eta_m, \text{ kN×m}.$$

3.4.4 Uneven rotation of the crankshaft

It should be noted that shown in Fig. 3.62 the average value of M_{ave} in the form of a straight line is purely theoretical, necessary for estimated calculations of the characteristics of energy consumers. In a real engine, due to the periodic action of tangential forces T from individual cylinders on the crankshaft, the change in the total tangential force ΣT, and therefore the torque, is periodic. This manifests itself in uneven rotation of the crankshaft. The force ΣT changes relative to a certain average value T_Σ (Fig. 3.63) with a period of change equal to the wedging angle of the cranks θ. Similarly, with a shift of half a period, the angular velocity ω of the crankshaft also changes.

For preliminary review

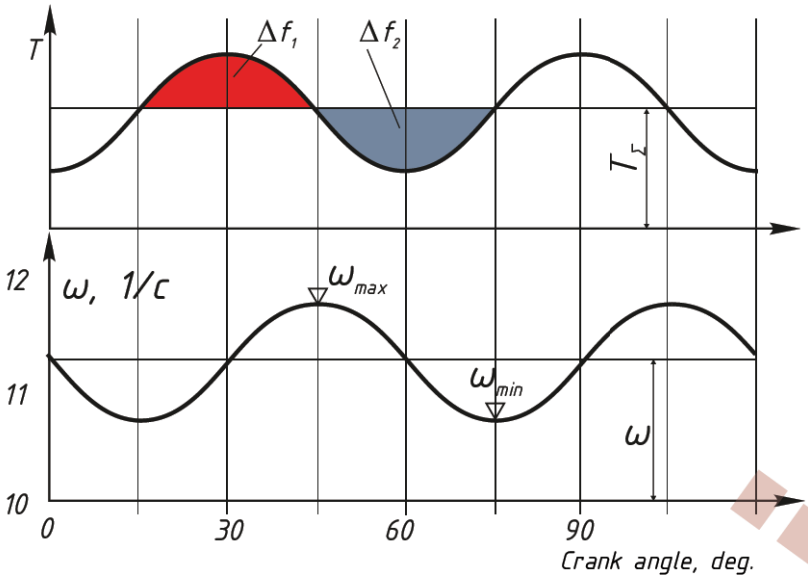


Figure 3.63 – Character of changes in the total tangential force T_Σ and angular velocity ω of rotation of the crankshaft of a 6-cylinder diesel engine

In relation to the average value of the torque, its actual change has characteristic sections (Fig. 3.63) in which its values are either excessive, which leads to a decrease in the angular velocity of the shaft, ωor insufficient, which leads to an increase in the angular velocity ω. In accordance with this, the work of the engine torque is sometimes greater than its average value, sometimes less. So, in the area Δf_1 work is negative ($-\Delta A$). The lack of this work is compensated by using part of the kinetic energy reserve of the rotating masses of the engine and connected energy consumers. In the next section Δf_2 the work is positive ($+\Delta A$). Excess work is spent on increasing the reserve of kinetic energy of the rotating masses, and the angular velocity increases.

Phenomena associated with the frequency of cylinder operation can be assessed numerically by an indicator – the degree of uneven rotation of the crankshaft δ:

$$\delta = (\omega_{max} - \omega_{min})/\omega,$$

where $\omega = (\omega_{max} + \omega_{min})/2$ – average angular speed of rotation of the crankshaft.

Based on the theorem on the change in the kinetic energy of the system, we write:

$$\Delta A = M\,(\omega^2_{max} - \omega^2_{min})/2,$$

where ΔA is the excess (or lack) of the work of the driving forces compared to the work of the resistance forces; M is the moment of inertia of the rotating masses of the shafting.

The value ΔA on the appropriate scale is determined by the area $\Delta f_1 = \Delta f_2$.

The degree of uneven rotation of the crankshaft for marine engines is in the range: δ = 1/200...1/300 for diesel generators and δ = 1/30...1/40 for main marine diesel engines (without taking into account the rotating masses of the shaft line and screw). The shaft of an engine with more cylinders rotates more evenly.

In small and medium-sized engines, the moment of inertia depends mainly on the mass of the flywheel, which determines the reserve of kinetic energy. Therefore, the required degree of uniformity of rotation of the crankshaft is ensured by selecting the appropriate flywheel parameters.

In low-speed marine engines, the flywheel has virtually no effect on the uniform rotation of the crankshaft – it is used as a drive gear for the rotating shaft and a connecting flange for the shaft line. When operating at low loads of the main engines, as well as when the cylinders are turned off in emergency situations, the unevenness of rotation increases by 2…3 times.

3.4.5 External imbalance and balancing methods for marine engines

The action in the crank mechanism of unbalanced periodically changing forces and their moments, which are transmitted to the housing parts of the engine, causes their vibration, which is transmitted to the ship's foundation and then to the ship's hull. The external manifestation of unbalanced forces is called external imbalance of the engine.

The forces from the gas pressure P_g, although they periodically change in direction, their action

For preliminary review

within each cylinder is mutually balanced, therefore these forces are not transmitted beyond the engine frame. The cause of engine imbalance is the inertia forces of the reduced translationally moving masses P_j and the unbalanced rotating masses of the cranks of all cylinders, as well as the moments created by these forces.

Inertial forces are divided into inertial forces of translationally moving masses, which include the piston, rod and part of the connecting rod, and centrifugal forces created by rotating masses.

The inertial force of translationally moving masses is described by the formula:

$$P_j = m_j r \omega^2 (\cos\varphi + \lambda_{cr} \cos 2\varphi). \quad (3.24)$$

As has already been shown above, based on the complex nature of its occurrence, the inertial forces of translationally moving masses can be represented as the sum of two components:

$$P_j = P_{j\text{I}} + P_{j\text{II}},$$

where

$$P_{j\text{I}} = m_j r \omega^2 \cos\varphi = P_{\text{I}} \cos\varphi, \quad (3.25)$$

called first-order inertia forces, which obey the cosine law of the crank angle and

$$P_{j\text{II}} = m_j r \omega^2 \lambda_{cr} \cos 2\varphi = P_{\text{II}} \cos 2\varphi, \quad (3.26)$$

called second-order inertial forces, which obey the cosine law of twice the crank angle of rotation 2φ.

The forces $P_{j\text{I}}$ and $P_{j\text{II}}$ act in the direction of motion of the translationally moving masses along the cylinder axis and are applied to the axis of the crank head connection (Fig. 3.64). The values of the second-order inertia forces are significantly less than the values of the first-order forces, since expression (3.26) contains $\lambda_{cr} = 0.2...0.5$.

The external influence of inertial forces of the first and second orders of progressively moving masses is manifested in the fact that, being directed upward, they tend to tear the engine away from the foundation, and when directed downward, press it to the foundation. The frequency of the forces and the vibrations they cause will be a multiple of ω, respectively 2ω.

Inertia force of rotating masses K (3.23) at a steady rotation speed is constant in value, applied at the center of the connecting rod journal and directed along the radius of the crank from the center of rotation (Fig. 3.64). Depending on the position of the crank, the inertial force of the rotating masses tends to displace the engine from the foundation in planes that coincide at each moment with the plane of the crank and passing through the axis of the crankshaft. The force vector K, lying in the plane of the crank, will rotate, changing its position in accordance with the change in angle φ.

In the plane of the crank (plane V), the force K can be transferred along the radius of the crank to the axis of the crankshaft O the form of the radius vector as K'.

The force K', without disturbing its action, can be transferred parallel to itself to any point along the axis of the crankshaft by attaching to it a pair of forces K'' and K''' with a moment equal to the moment of force K' relative to point O_1 (Fig. 3.64 *a*). Using this rule, we bring the force K to the center O_s, which is the point of intersection of the crankshaft axis with the plane S in which the center of gravity of the engine lies. In this case, the force K''' is the reaction of the support to the force K''.

After reduction we have:

- force K applied to the center O and tending to tear the engine away from the foundation in the plane V;
- a pair of forces K' and K''', creating a moment M_k on arm l_1, acting in the same plane V and tending to tilt the engine relative to its center of gravity in a clockwise direction.

The resulting moment is called the moment of centrifugal force, it is equal to:

$$M_k = Kl_1 = l_1 m_R r \omega^2. \quad (3.27)$$

The magnitude of this moment and the direction of its action depend both on the magnitude of the force K and on the position of the plane passing through the center of gravity of the engine relative to the cylinder axis.

For preliminary review

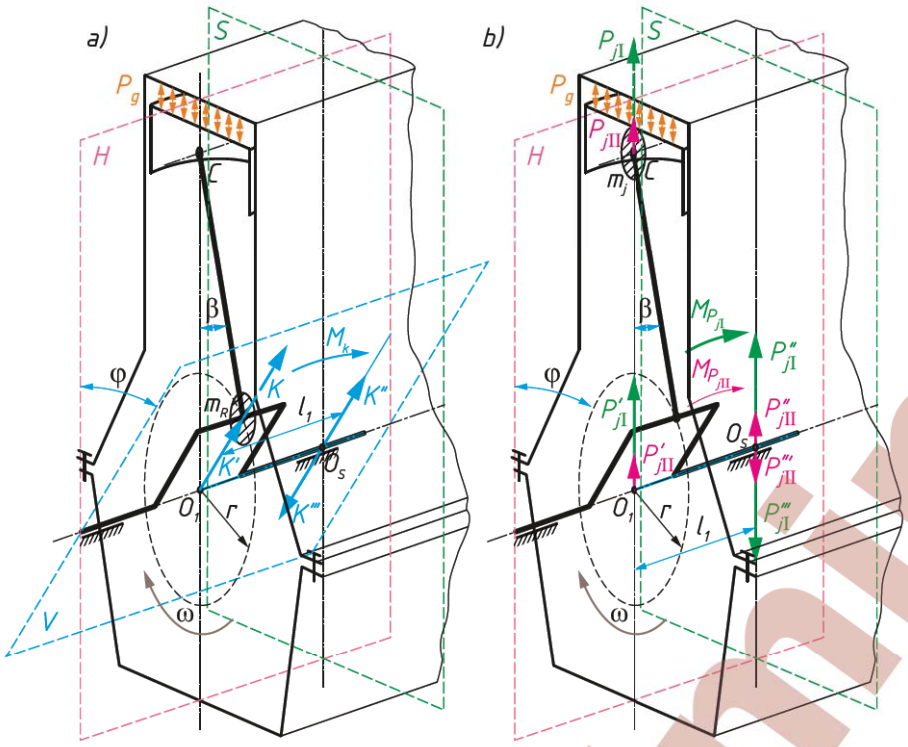


Figure 3.64 – Inertial forces of masses performing rotational motion and the moments they create (*a*); inertial forces of translationally moving masses and the moment they create (*b*)

The moment of centrifugal force M_k, as well as the force K itself, when the shaft rotates, remain unchanged and each time, acting in the plane of the knee, they rotate with it at an angular speed ω.

Similarly (Fig. 3.09 *b*) moments from the forces of inertia of masses moving back and forth, first order (forces P'_{j_I} and P'''_{j_I} on shoulder l_1):

$$M_{P_{j_I}} = l_1 P_{j_I} = l_1 m_j r \omega^2 \cos\varphi$$

and second order (forces $P'_{j_{II}}$ and $P'''_{j_{II}}$ on shoulder l_1):

$$M_{P_{j_{II}}} = l_1 P_{j_{II}} = l_1 m_j r \omega^2 \lambda_{cr} \cos 2\varphi .$$

Since the forces P_{jI} and P_{jII} act in the plane of motion of the piston, the moments $M_{P_{j_I}}$ and $M_{P_{j_{II}}}$ do not depend on the position of the crank and act in the plane H (Fig. 3.64) passing through the cylinder axis. The external influence of these moments tends to tilt the engine relative to its center of gravity in a vertical plane passing through the cylinder axes.

External imbalance of a multi-cylinder diesel engine. External imbalance of the engine means the presence in it of periodic forces or moments of force transmitted to the foundation, which cause vibration. The forces and moments discussed above act within each cylinder. In a multi-cylinder engine, the total effect of the forces and moments acting in each cylinder can be obtained by finding their vector sums:

$$\sum_1^i K;\ \sum_1^i P_{j_I};\ \sum_1^i P_{j_{II}};\ \sum_1^i M_k;\ \sum_1^i M_{P_{j_{II}}};\ \sum_1^i M_{P_{j_{II}}};$$

where i is the number of engine cylinders.

If, as a result of the summation of the acting forces and moments, their vector sum is equal to zero, the external influence of the resulting vectors of forces and moments in this case is absent and they are not transmitted to the foundation of the vessel. In this case, the engine is called completely externally balanced.

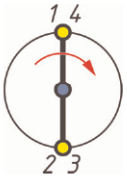
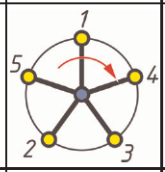
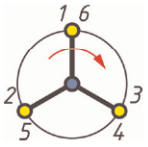
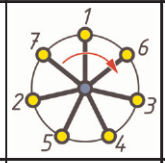
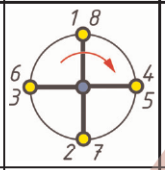
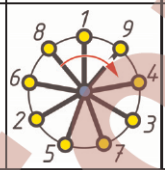
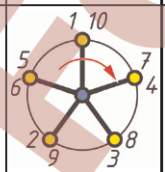
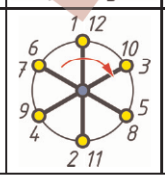
Zero values of vector sums can be obtained if the forces of the same name acting in all cylinders, and the moments of these forces, are closed inside the engine frame, compensating each other.

If the vectors of forces or moments during addition are not compensated or are partially compen-

For preliminary review

sated, the resulting vectors will go beyond the engine frame and have an external periodic effect on the foundation of the vessel. In this case, the engine is considered externally unbalanced. Since the geometric sum of vectors depends on their number, size and relative position, the external imbalance of a multi-cylinder engine is generally determined by the engine clock cycle, the number of cylinders, and the wedging angle of the cranks and the operating order of the cylinders. The most typical cases of these factors characteristic of marine engines are given in table. 3.1-3.2.

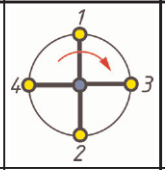
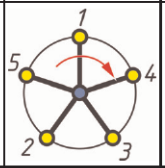
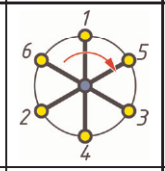
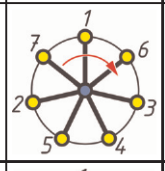
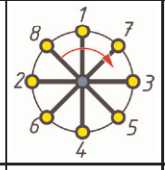
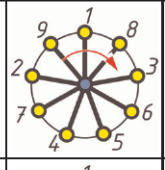
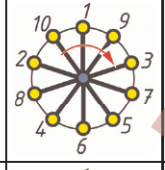
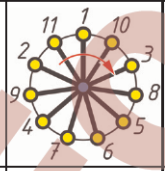
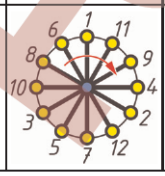
Table 3.1 – Crank layouts and maximum values of unbalanced forces and moments of single-row four-stroke engines

Number of cylinders	Crank diagram	Flash sequence	Crank wedge angle	Cylinder operating order	Resultant inertial forces and moments of inertial forces					
					$\sum_1^i K$	$\sum_1^i P_{j_I}$	$\sum_1^i P_{j_{II}}$	$\sum_1^i M_k$	$\sum_1^i M_{P_{j_I}}$	$\sum_1^i M_{P_{j_{II}}}$
4		180°	180°	1-3-4-2 1-2-4-3	0	0	$4P_{j_{II}}$	0	0	0
5		72°	144°	1-2-4-5-3	0	0	0	$0.449\,Ka$	$0.449\,P_{jI}a$	$4.98\,P_{jII}a$
6		120°	120°	1-5-3-6-2-4 1-4-1-6-3-5 1-3-5-6-4-2	0	0	0	0	0	0
7		51°26′	102°52′	1-2-4-6-7-5-3	0	0	0	$0.276\,Ka$	$0.276\,P_{jI}a$	$1.006\,P_{jII}a$
8		90°	90°	1-6-2-5-8-3-7-4 1-6-2-4-8-3-7-5 1-3-7-5-8-6-2-4 1-3-7-4-8-6-2-5	0	0	0	0	0	0
9		40°	80°	1-8-5-3-9- -6-2-7-4	0	0	0	$0.92\,Ka$	$0.92\,P_{jI}a$	$1.8\,P_{jII}a$
10		72°	72°	1-6-2-8-4- -10-5-9-3-7 1-6-9-3-7- -10-5-2-8-4	0	0	0	0	0	0
12		60°	60°	1-6-9-2-8-3-12- -7-4-11-5-10	0	0	0	0	0	0

a – distance between the axes of the working cylinders

For preliminary review

Table 3.2 – Crank layouts and maximum values of unbalanced forces and moments of single-row two-stroke engines

Number of cylinders	Crank diagram	Flash sequence	Crank wedge angle	Cylinder operating order	Resultant inertial forces and moments of inertial forces					
					$\sum_1^i K$	$\sum_1^i P_{j_I}$	$\sum_1^i P_{j_{II}}$	$\sum_1^i M_k$	$\sum_1^i M_{P_{j_I}}$	$\sum_1^i M_{P_{j_{II}}}$
4		90°	90°	1-4-2-3 1-3-2-4	0	0	0	$1.41\,Ka$	$1.41\,P_{jI}a$	$4.0\,P_{jII}a$
5		72°	72°	1-5-2-3-4	0	0	0	$0.449\,Ka$	$0.449\,P_{jI}a$	$4.98\,P_{jII}a$
6		60°	60°	1-6-2-4-3-5 1-5-3-4-2-6	0	0	0	0	0	$3.464\,P_{jII}a$
7		51°26′	51°26′	1-7-2-5-4-3-6	0	0	0	$0.276\,Ka$	$0.276\,P_{jI}a$	$1.006\,P_{jII}a$
8		45°	45°	1-8-2-6-4- -5-3-7	0	0	0	$0.448\,Ka$	$0.448\,P_{jI}a$	0
9		40°	40°	1-9-2-7- -4-5-6-3-8	0	0	0	$0.194\,Ka$	$0.194\,P_{jI}a$	$0.548\,P_{jII}a$
10		36°	36°	1-10-2-8-4- -6-5-7-3-9	0	0	0	0	0	$0.898\,P_{jII}a$
11		32°44′	32°44′	1-11-2-9-4-7- -6-5-8-3-10	0	0	0	$0.153\,Ka$	$0.153\,P_{jI}a$	$0.382\,P_{jII}a$
12		30°	30°	1-6-8-10-3-5- -7-12-2-4-9-11	0	0	0	0	0	0

a – distance between the axes of the working cylinders

Engine balancing refers to design solutions aimed at reducing the effect of external forces and moments.

For preliminary review

In marine internal combustion engines, the centrifugal forces of inertia of the rotating masses, the inertial forces of the first and second order, as well as the moments caused by these forces, are balanced.

First-order inertia forces arise as a result of changes in the directions of movement of the piston group. These forces reach peak values when the piston moves at dead points. The consequence of the occurrence of first-order forces is transverse engine vibration, the frequency of which is equal to the crankshaft rotation speed. Typically these forces are partially balanced by balancers mounted on the crankshaft. Complete balancing of first-order inertial forces in this way is impossible, since the balancers themselves perform a rotational movement, and the balanced parts of the piston group are reciprocating.

Second-order inertia forces arise as a result of changes in the magnitude of the linear speed of movement of the piston in the process of moving it between dead points. These forces reach their maximum value in the middle of the piston stroke and cause transverse engine vibration, the frequency of which is twice the crankshaft speed. Second-order inertia forces are very difficult to balance, and since their magnitude is 2...4 times less than the first-order inertia forces, most often the second-order forces are left unbalanced so as not to complicate the engine design.

In addition to the external influence of inertial forces and their moments, the latter, acting inside the engine frame, load and deform its structural elements. Centrifugal forces, acting in the plane of the crankshaft, and inertial forces of the first and second orders in the plane of the cylinder axes, load the crankshaft and bearings and are transmitted to the foundation frame.

At the same time, the shaft is loaded by moments M_k, $M_{P_{J_I}}$ and $M_{P_{J_{II}}}$, which tend to bend the shaft in the planes of their action. The deformation of the shaft is perceived by the bearings (especially the central ones, which experience the greatest load) and the foundation frame in which they are located. As a result, bending and deformation stresses arise in the foundation frame, as well as on the shaft, under the influence of moments of inertia.

The noted action of inertial forces inside the engine frame determines the internal imbalance of the engine

It should be noted that regardless of the degree of external imbalance and the nature of its manifestation, the internal engine always remains unbalanced. The higher the rotational speed of the crankshaft and the greater the mass of the moving parts, the greater the inertial force and the greater the effect of their internal and external imbalance.

Balancing methods. External imbalance of the engine, especially if the unbalanced forces and moments are significant, leads to vibrations of both the engine itself and the ship's hull structures connected to it. Therefore, to reduce vibration, they resort to balancing the forces and moments occurring in the engine.

Balancing the centrifugal forces is carried out by installing counterweights on the crank cheeks (Fig. 3.65).

When rotating, the crank is acted upon by a centrifugal force K, lying in the plane of its rotation, which consists of two components: force K_1 from the mass of the connecting rod, attributed to the rotationally moving parts, force K_2 from the mass of the unbalanced part of the crankshaft.

The force K can be balanced by two counterweights mounted on the cheeks of the crank. They develop centrifugal force ($K^1_{cw} + K^2_{cw} \approx K$).

Then we can write:

$$m_R r\omega^2 = 2m_{cw} k\omega^2$$

or if the angular velocities of the engine masses and counterweights are equal

$$m_R r = 2\, m_{cw} k$$

where m_{cw} is the mass of one counterweight, kg; k – radius of inertia of the counterweight, equal to the distance from its center of gravity to the axis of rotation, m.

The centrifugal forces of inertia of rotating masses can be balanced in an engine with any number of cylinders by installing counterweights on the crankshaft. In most multi-cylinder engines, the

For preliminary review

resulting inertial forces are balanced not by installing counterweights, but by selecting the appropriate number and wedge angles of the crankshaft cranks. On most marine engines, it is impossible to balance the inertial forces of the first and second orders by installing counterweights. However, with an appropriate choice of the counterweight mass, it is possible to partially transfer the action of the first-order inertial force from one plane to another, thereby reducing the imbalance in this plane. Therefore, counterweights are installed even on balanced shafts to reduce and more evenly distribute the load on the frame journals and bearings, as well as to reduce the bending moments of the crankshaft.

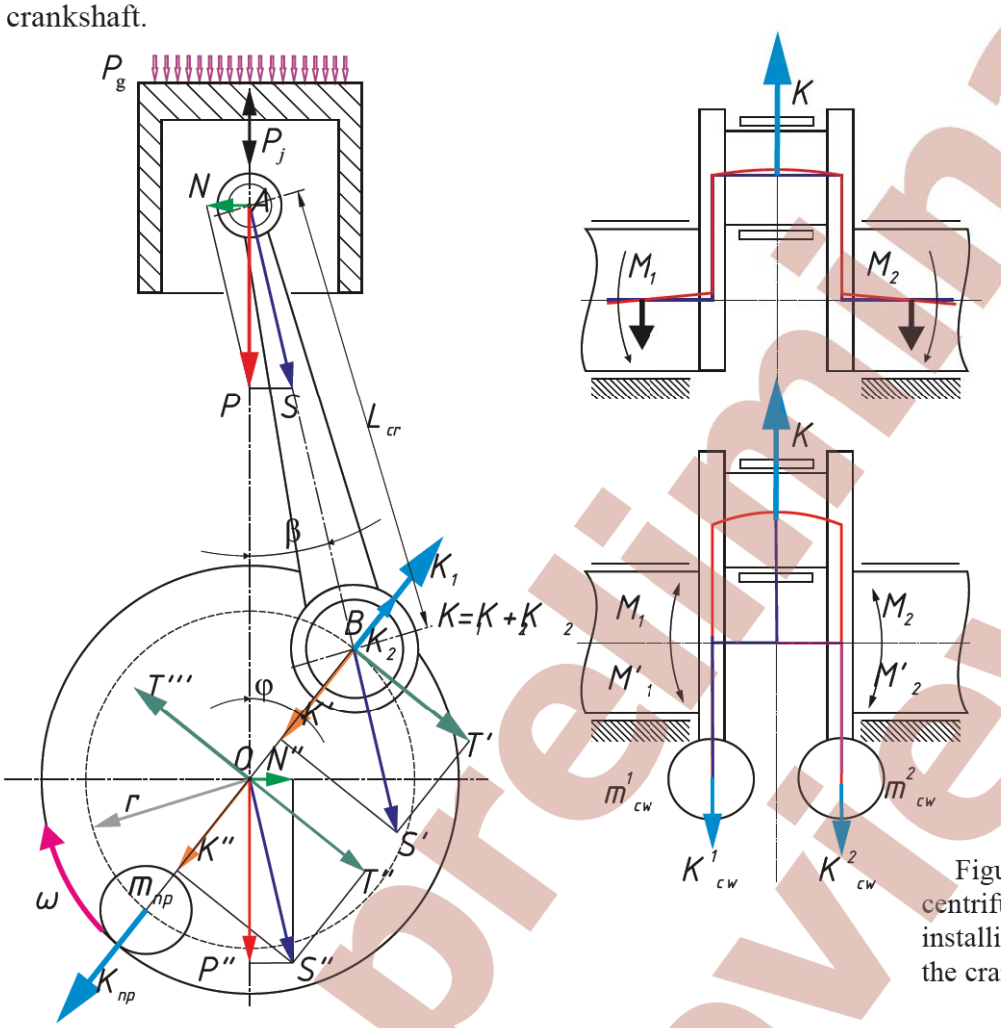


Figure 3.65 – Balancing centrifugal forces of inertia by installing counterweights on the crank cheeks

Unbalanced moments can be fully or partially compensated:

- by balancing centrifugal forces within each cylinder

- by balancing the free moment with a pair of counterweights that develop an equal but opposite moment. The counterweights are placed at the maximum possible distance from each other, which allows them to reduce their weight.

The first order moments for most two-stroke engines with standard balancing have the same values. For example, on four-cylinder two-stroke engines they act in both vertical and horizontal directions. Therefore, as standard, four-cylinder engines are equipped with adjustable counterweights (Fig. 3.66). They can significantly reduce the vertical moment (although they increase the horizontal moment, which is much easier to compensate). For engines with five cylinders or more, moments of the first order are not significant (Table 3.2) and do not affect the ship's hull

Sometimes they resort to installing a counterweight to unload the frame bearings. In this case, the counterweights are hung on the cheeks at an angle to their axis.

Balancing the inertial forces of the first and second orders arising from progressively moving masses cannot be achieved using counterweights installed on the cheeks of the shaft, since the centrifugal force of the counterweights, being in the plane of the knee, continuously changes the plane of its action when the shaft rotates.

For preliminary review

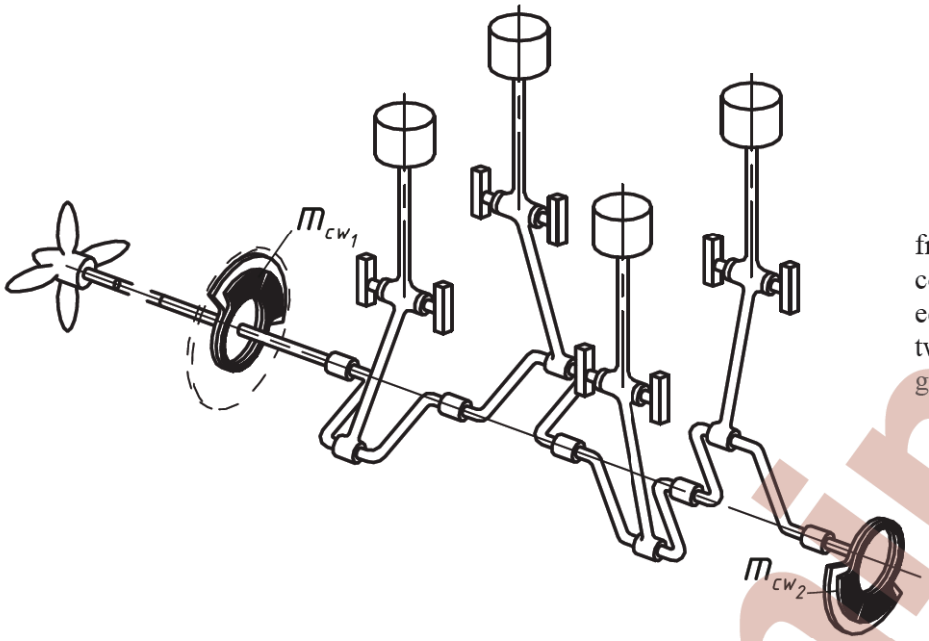


Figure 3.66 – Balancing free torque with a pair of counterweights developing equal but opposite torque in a two-stroke four-cylinder engine (adapted from [48])

In cases where a first-order moment causes resonance of the housing, both in the vertical and horizontal planes, in the operating range of engine speeds, the inertia forces are balanced by counterweights located on additional shafts that are mechanically connected to the crankshaft. Thus, on low-speed MAN engines, a first-order torque compensator can be additionally mounted, which allows them to be reduced to an acceptable value. To create a balancing moment, an oppositely directed force acting in the same plane is required. The compensator contains two counter-rotating weights mounted on chain tension sprockets and rotating at the same speed as the crankshaft (Fig. 3.67).

These masses are wedged on additional shafts in such a way that the components of the centrifugal forces created during their rotation add up and produce an oppositely directed force, which balances the first-order force.

In the same way, it is possible to balance the 2nd order inertial force, but for this it is necessary that the counterweights rotate in opposite directions with double the angular velocity 2ω. To do this, an additional compensator is installed at the front end of the engine (or at both ends), driven from the crankshaft through a separate chain drive with a rotation speed twice the engine speed (Fig. 3.68).

As can be seen from Table 3.2, two-stroke engines have unbalanced moments from first- and second-order forces, the values of which are proportional to the crank radius r and the ratio λ_{cr}. With an increase in the piston stroke of low-speed engines, the inertial forces and the moments they cause increase, which requires special measures to balance them. The task of balancing the engine is solved in conjunction with other tasks aimed at reducing vibration of the ship's hull as a whole.

The overturning moment is the only moment that cannot be balanced, but its unevenness (amplitude) can be reduced by reducing the unevenness of the torque. This is achieved by increasing the number of engine cylinders with equal intervals between flashes (power strokes) in them.

3.4.6 Torsional and axial vibrations

Since changes in torque occur periodically, and the crankshaft is not an absolutely rigid system and has certain elasticity, when maximum torque values are applied, it twists at a certain angle, and when the value of the applied torque decreases, it unwinds under the influence of elastic forces. Thus, periodic exposure to torque leads to twisting and unwinding of shaft sections with a frequency equal to the frequency of flashes in the engine cylinders. This behavior of the crankshaft system with attached masses is called *torsional vibration*.

Torsional vibrations are periodic vibrations in the plane of rotation of masses suspended on a shaft, in which sections of the shaft between the masses are twisted and untwisted under the influ-

For preliminary review

ence of cyclically varying torque and inertia forces of the crank mechanism.

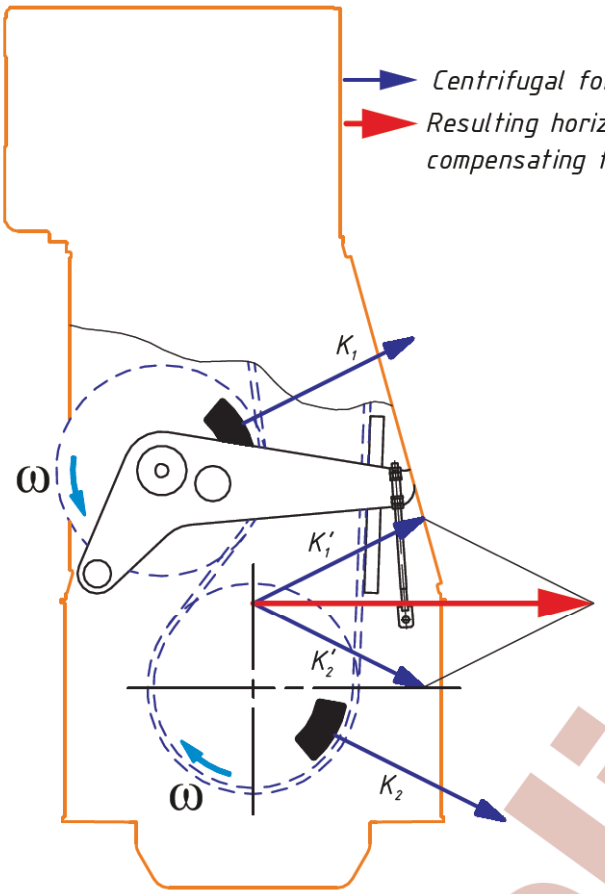


Figure 3.67 – First order moment compensator containing two counter-rotating weights mounted on chain sprockets and rotating at the same frequency as the crankshaft, seen from aft (adapted from [48])

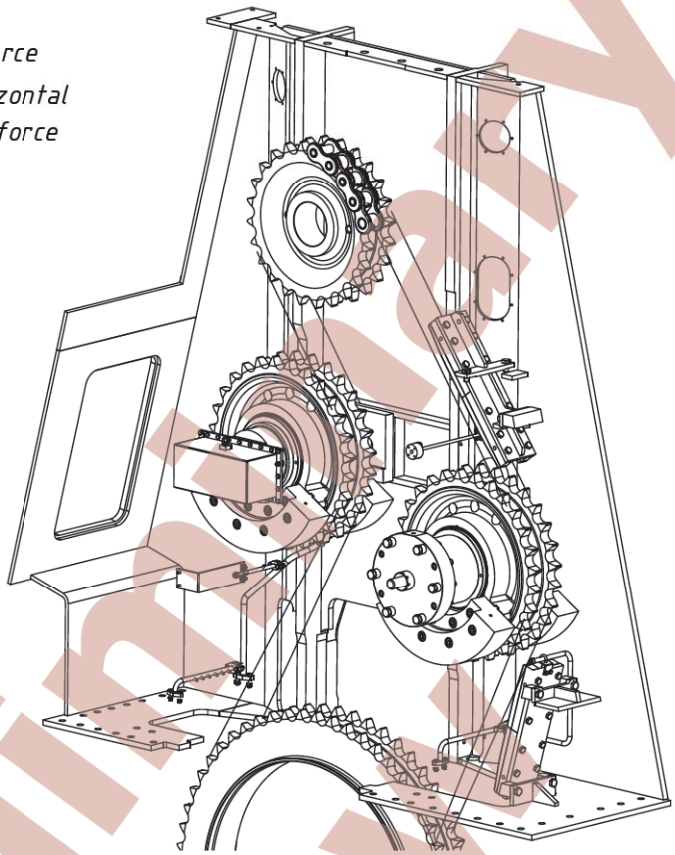


Figure 3.68 – Second order moment compensator containing two counter-rotating weights mounted on sprockets driven from the crankshaft with a frequency twice the engine speed (adapted from [49])

The nature of the occurrence and action of torsional vibrations is influenced not only by the engine's own elastic elements, but also by the power transmission elements and energy consumers attached to it. The attached masses include: a flywheel, intermediate and propeller shafts with masses suspended on them, couplings, a propeller and other elements connected into a single elastic system. Thus, the engine crankshaft and the ship's shaft line form an oscillatory system, which includes two dissimilar energy sources: the elasticity of the shafts (potential energy) and mass (kinetic energy). When the external influence on the shaft line changes, the potential energy of the twisted shaft is converted into the kinetic energy of oscillating masses and vice versa. Torsional vibrations can lead to large dynamic stresses, and when they accumulate, to the destruction of system elements.

Any oscillatory system under the influence of alternating forces experiences two types of oscillations – free and forced.

Free vibrations using the example of the simplest torsional system, consisting of a shaft of length l rigidly fixed at one end, and a mass in the form of a disk, the center of gravity of which lies on the axis of the shaft (Fig. 3.69 *a*). If a moment M_{tor} is applied to the disk, then the shaft, within the limits of elastic deformation, will twist at an angle $+\varphi$. In this case, the elastic forces of the material counteract torsion with a moment M_{res}. After the termination of the moment M_{tor}, under the influence of the moment M_{res}, the system will return to its original position, however, under the influence of inertial forces, the disk will not stop in the equilibrium position, but will continue to twist the shaft in the opposite direction at an angle $-\varphi$. After reaching the maximum angle of twist, the elastic forces of the shaft will again cause the disk to rotate, and by inertia it will pass through the

For preliminary review

equilibrium position. Thus, after the cessation of the external moment, the system, due to the elastic forces of the shaft and inertial forces, will perform free oscillations during which the shaft will twist in one direction or the other.

If the rigid fastening of the free end of the shaft is replaced with significant resistance, for example, by installing a propeller or a generator flange at the end of the shaft, the loaded end of the shaft will rotate uniformly, and the free end will oscillate.

Reviewed in Fig. 3.69 and the torsional system is the simplest single-mass system. If another disk is attached to the free end of the shaft (Fig. 3.69 *b*), we obtain a two-mass torsional system. If to disks masses m_1 and m_2 apply moments M_{tor} and M_{res} of opposite signs, the shaft will spin in opposite directions. When the action of the moments ceases, elastic vibrations will occur relative to a certain section in which the angle of twist will be equal to zero. This section of the shaft is called the oscillation node. If on a scale diagram we plot the maximum values of the angular amplitudes of vibrations and connect the resulting points with a straight line, we will obtain a graph of the angular amplitude deviations of masses from the equilibrium position (Fig. 3.69 *b*). A two-mass system has only one oscillation node in the *g* plane, and is therefore called a single-node system. The actual torsional system of a ship's propulsion installation is a complex multi-mass system in which the number of oscillation nodes is less than the number of masses per unit. In practice, when determining the frequency of free oscillations, the propulsion installation circuit is simplified and leads to a three-mass system: engine-flywheel-propeller (Fig. 3.69 *c*). Such a system can have both single-node and two-node vibration modes, each of which has its own free vibration frequency. The more nodes in the system, the higher the frequency of its oscillations.

Forced oscillations in the shafting system arise under the influence of periodically changing torques from gas pressure forces in the cylinders and inertia forces of progressively moving masses. These moments are called disturbing moments.

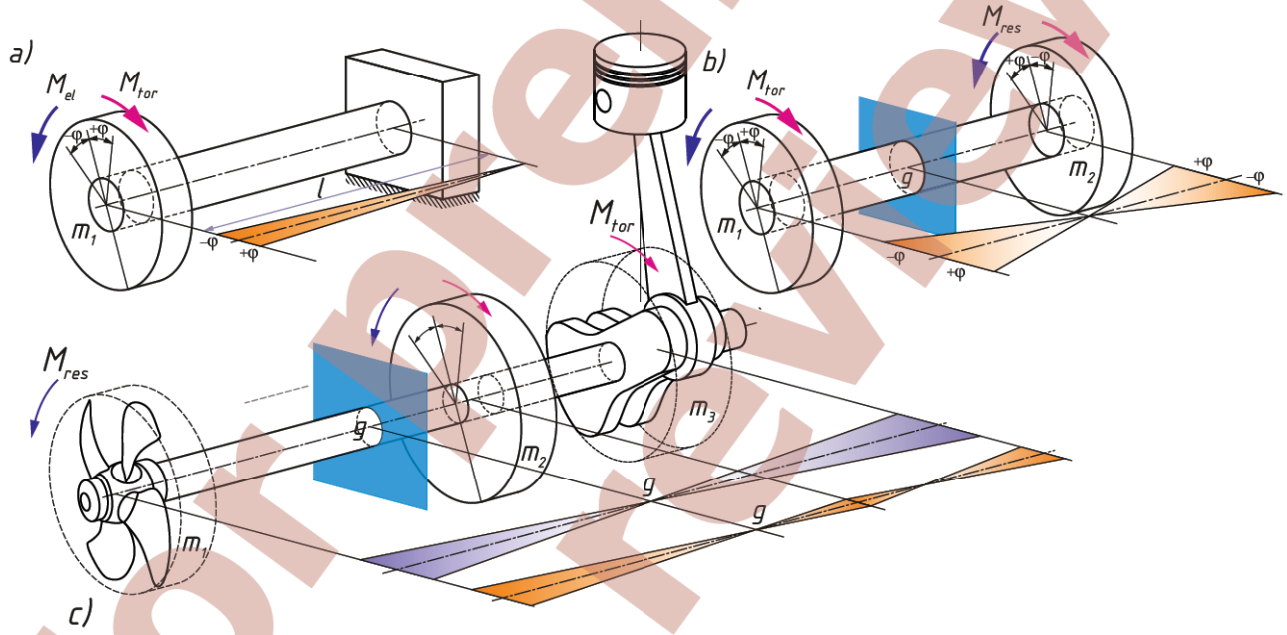


Figure 3.69 – Torsional systems: *a* – single-mass; *b* – two-mass; *c* – three-mass

Forced oscillations in the shafting system arise under the influence of periodically changing torques from gas pressure forces in the cylinders and inertia forces of progressively moving masses. These moments are called disturbing moments.

As has already been shown above, the forces P_{jI} and P_{jII} and the moments they create can be decomposed into elementary components – harmonics, varying with different amplitudes and frequencies. This decomposition is called harmonic analysis.

Each harmonic has its own order, which shows the number of its complete changes per one revolution of the shaft. Each harmonic excites forced oscillations in the shaft of a certain frequency,

For preliminary review

proportional to the order of the harmonic and the shaft rotation frequency. The largest amplitudes of forced torsional vibrations are caused by harmonics, the order of which is equal to or a multiple of the number of flashes in the cylinders per revolution of the engine shaft. They are called the main harmonics, and their smallest order is $k = i/z$, where i is the number of cylinders; z – engine stroke coefficient. The remaining orders of the main harmonics will be multiples of it. For example, for a six-cylinder four-stroke engine $k = 6/2 = 3$ (multiple orders: 6, 9, 12, etc.), for a two-stroke engine $k = 6/1 = 6$ (multiple orders: 12, 18, 24, etc.d.).

Thus, forced torsional vibrations under the influence of torque can be considered as the sum of harmonic vibrations under the influence of individual components of this moment.

Resonance, critical speed and exclusion zones. The coincidence of the frequency of any form of free oscillation with the frequency of any harmonic component of the disturbing moment is called resonance, and the rotation frequency at which resonance occurs is called the critical or resonant frequency. During resonance, the amplitude of torsional vibrations (the angle of twist of the shaft) and the additional torsional stresses proportional to it increase and can be dangerous for the strength of the shaft and cause its failure.

Rotation speed ranges near critical frequencies are dangerous for long-term engine operation, and are therefore called critical. They are marked on tachometer dials as prohibited zones, long-term work in which is unacceptable, and their passage must occur as quickly as possible.

If the restricted rotation speed zone is in the area of engine operating conditions, measures are taken to shift outside this area. To do this, even during the design process of the installation, the frequency of free vibrations of the system is changed by installing special torsional vibration dampers – dampers.

Torsional vibration dampers are used to reduce the amplitude of torsional vibrations of the crankshaft. They are usually installed at the free end of the crankshaft, or less commonly the camshaft. Their presence creates favorable conditions for the operation of both the crankshaft itself and the operation of auxiliary mechanism drives, ensuring constant angular velocity.

According to the method of influencing the crankshaft system, dampers are divided into antivibrators and torsional vibration dampers.

Antivibrators operate without absorbing the vibrational energy of the system. The most common are pendulum antivibrators with bifilar pendulum suspension. They work on the principle of "detuning» the resonance system of the shaft, so they are effective only in a narrow range of engine speeds. In other modes they are ineffective. For this reason, antivibrators are used in engines of diesel generator sets operating at constant speed.

Dampers are dampers with partial absorption and subsequent dissipation of torsional vibration energy into the environment.

Absorption of vibration energy is carried out either by increasing dry friction in the system (dry friction dampers), or due to liquid friction (liquid friction dampers), or due to intramolecular friction in the material of the elastic element (rubber dampers). Dampers effectively reduce the amplitude of torsional vibrations in a wide range of engine speeds, including resonant ones. Therefore, dampers are widely used on modern marine diesel engines.

On LSEs and large MSEs liquid vibration dampers are used; on MSEs and HSEs, dry friction dampers are used; on HSEs of relatively low power, rubber dampers are sometimes used.

The fluid friction damper is an annular housing attached to a flange at the free end of the crankshaft or camshaft (Fig. 3.70). The housing houses a massive annular flywheel that rotates freely relative to the housing.

The gaps between the housing and the flywheel are filled with a special silicone liquid with high viscosity.

When rotating, the flywheel housing performs oscillatory movements together with the crankshaft, while the flywheel, which is not mechanically connected to the housing, strives to maintain a constant angular velocity due to inertial forces. When a difference in speed between the housing and the flywheel occurs, friction between the layers occurs in the liquid, during which the energy of torsional vibrations is converted into thermal energy. In this case, the amplitude of oscillations is sig-

For preliminary review

nificantly reduced.

To effectively remove heat from the damper housing, it is usually located outside the engine crankcase.

The gaps between the housing and the flywheel are filled with a special silicone liquid with high viscosity.

When rotating, the flywheel housing performs oscillatory movements together with the crankshaft, while the flywheel, which is not mechanically connected to the housing, strives to maintain a constant angular velocity due to inertia forces. When a difference in speed between the housing and the flywheel occurs, friction between the layers occurs in the liquid, during which the energy of torsional vibrations is converted into thermal energy. In this case, the amplitude of oscillations is significantly reduced.

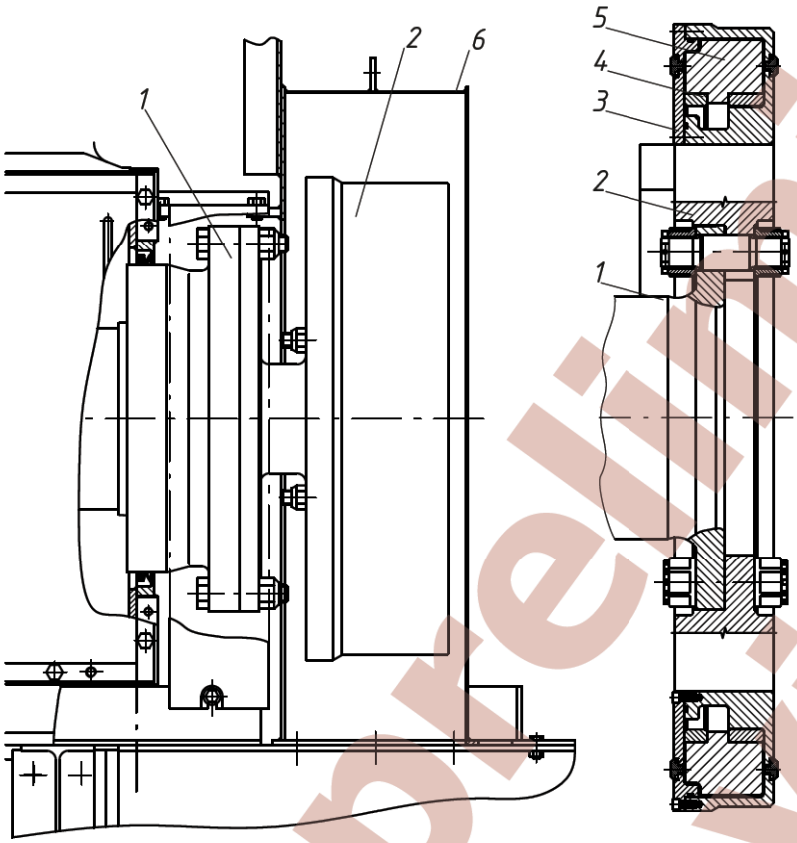


Figure 3.70 – Liquid torsional vibration damper: 1 – shaft flange; 2 – damper housing; 3 – housing cover; 4 – flywheel bushing; 5 – flywheel; 6 – protective casing (adapted from [18])

To effectively remove heat from the damper housing, it is usually located outside the engine crankcase.

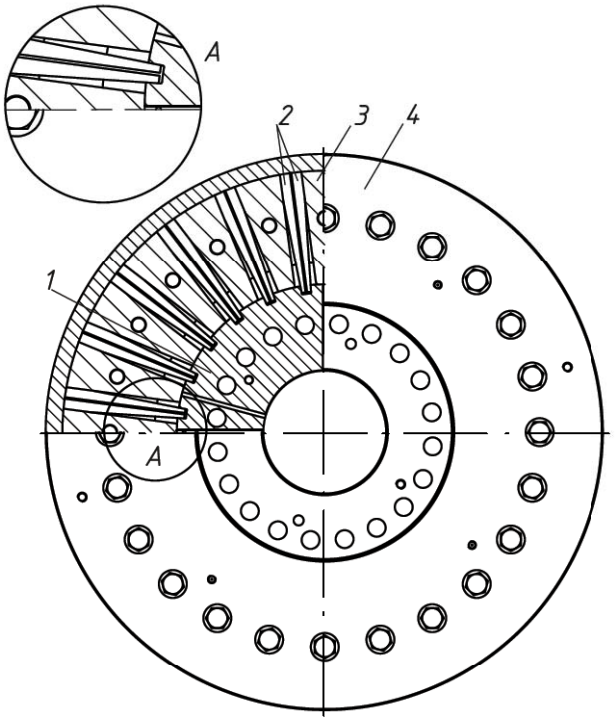
The dry friction damper consists of a hub rigidly mounted on the crankshaft or camshaft of the engine. A stacked stator is put on the hub, which consists of a housing, wedge-shaped segments and flat paired springs sandwiched between them (Fig. 3.71).

The springs have gentle bevels, the ends of which fit into the slots on the hub. After assembly, the wedge-shaped segments are rigidly connected to the housing and cover, forming a flywheel elastically connected to the hub.

Torsional vibrations of the shaft lead to relative movement of the hub and flywheel, which causes elastic deformation of the springs. In this case, the energy of torsional vibrations is spent on overcoming the elastic forces of the springs, and the amplitude of torsional vibrations decreases.

In addition to torsional vibrations in the crankshaft system, longitudinal vibrations also occur along the axis of rotation of the shaft. The nature of these vibrations is associated with deformation of the knees in the form of alternating divergence or convergence of the cheeks under the influence of forces loading the cranks. As a result, the shaft acquires vibrations along the axis, which are transmitted to the thrust bearing, and through it to the foundation and hull of the vessel. With large crankshaft sizes, axial vibrations can reach significant values. To reduce the amplitude of axial vibrations, in some cases an axial vibration damper is installed at the free end of the shaft.

For preliminary review



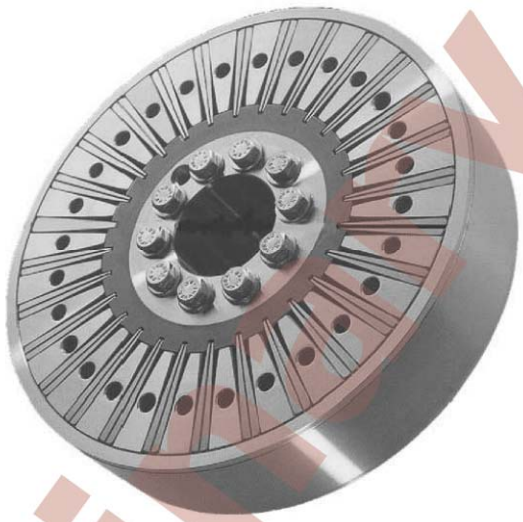


Figure 3.71 – Dry friction torsional vibration damper: 1 – crankshaft hub; 2 – flat springs; 3 – stator segments; 4 – housing cover (adapted from [50])

Axial vibration damper usually installed at the free end of the crankshaft. An example of the design of such a damper for a low-speed marine engine is shown in Fig. 3.72.

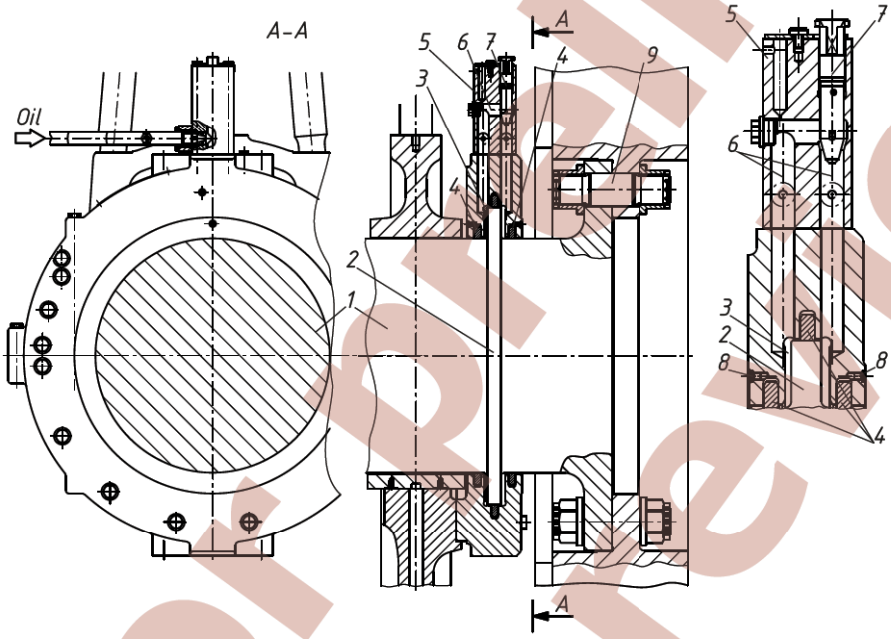


Figure 3.72 – Damper of axial vibrations of the crankshaft of the low-speed RTA 72U engine: 1 – engine crankshaft journal; 2 – annular protrusion; 3 – hydraulic chamber; 4 – seals of hydraulic cavities; 5 – throttle device; 6 – U-shaped channel; 7 – conical locking device; 8 – drainage holes for draining oil; 9 – torsional vibration damper mounting flange (adapted from [18])

There is an annular protrusion on the shaft neck, which plays the role of a hydraulic piston. On both sides of the protrusion there is a hydraulic chamber, consisting of two halves and attached to the end of the diesel engine frame. The protrusion divides the chamber volume into two cavities, into which oil from the diesel lubrication system flows through special channels. To seal the cavities, three o-ring seals are installed in the housing of the hydraulic chamber, two seal the gaps between the chamber and the shaft and one seals the gap between the protrusion and the chamber. During engine operation, axial vibrations of the shaft cause movement of the protrusion. In this case, the oil is forced out from one cavity to another through a U-shaped channel in which an adjustable throttle with a cone locking device is installed. The required damping energy is created due to hydraulic resistance when oil flows through the gap in the throttle. The position of the cone valve can be adjusted, which allows you to change the characteristics of the damper. To remove the heat

For preliminary review

generated during damper operation, continuous circulation is provided – the oil is constantly drained through drainage holes in the chamber housing. The diameter of the holes is relatively small and they do not affect the operation of the device.

Literature used in writing section 3:

1. Operating Instructions TBD 645. 0299 6391 En, Motoren-Werke Mannheim AG, Mannheim. Germany. – 244 p.
2. Engine Type: 8K90MC-C. Index for Description Manual. Vol.03 Description Manual. 912 Assembly of large parts. Doosan Engine Co. Ltd. Gyeongnam, Korea. – 14 p.
3. Spahni M., Brunner H. Wärtsilä 82 Low-Speed Engine Types. High-Economy Engines for Panamax Containerships, Large Tankers and Large Ore Carriers. Wärtsilä Switzerland Ltd, Winterthur. 2008. – 15 p.
4. Geist M. Sulzer RTA-8T engines: Compact two-strokes for tankers and bulk carriers. Wärtsilä NSD Switzerland Ltd, Winterthur. 1998. – 53 p.
5. L16/24. Section 519. Fitting Instructions for Resilient Mounting of GenSets, 519-03.00. MAN B&W Diesel A/S Holeby. 1994 – 4 p.
6. Technology review Wärtsilä 46F engine. Wärtsilä Corporation. Helsinki. 2004 – 20 p.
7. Presentation of Service Experience with Medium and Large Bore MC Engines. MAN B&W Diesel A/S. 2002 – 124 p.
8. Demmerle R. Sulzer RTA84C and RTA96C engines: The reliable driving forces for large, fast containerships. Wärtsilä NSD Switzerland Ltd, Winterthur. 1997. – 51 p.
9. Engine Type: 7S70ME-C. Index for Description Manual. Vol.03 Description Manual. 912 Assembly of large parts. Doosan Engine Co. Ltd. Gyeongnam, Korea. – 14 p.
10. Sulzer RTA-T. Technology Review. Wärtsilä Corporation. Helsinki – 24 p.
11. L23/30H. Section 511. Frame with Main Bearings. 51101-10H. MAN B&W Diesel A/S. Holeby Generatinc Sets. 1994. – 2 p.
12. Engine operating instructions. Type M43C. Caterpillar Motoren GmbH & Co. KG. 2007 – 2351 p.
13. Engine Type: 8K90MC-C. Index for Description Manual. Vol.03 Description Manual. Section 903. Cyinder Liner and Cylinder Lubrication. HSD Engine CO. LTD. Korea. – 14 p.
14. 6S90MC-C and 7K98MC on Testbed. MAN B&W Diesel A/S. 2000 – 56 p.
15. 6L21/31 Vol. 1, Index for Maintenance Manual. MAN Diesel & Turbo. 2015 – 460 p.
16. Wärtsilä 46 C. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 421 p.
17. Sulzer RT-flex82C Maintenance Manual «Marine». Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2008. – 143 p.
18. Sulzer RTA72U-B. Maintenance Manual «Marine». Wärtsilä Switzerland Ltd., Winterthur, Switzerland. 2000 – 396 p.
19. MAN Diesel. 50-108ME-ME-C. Vol. I. Operation. MAN B&W Diesel A/S. Copenhagen, Denmark. – 623 p.
20. Bearings. MAN B&W Diesel A/S. Copenhagen. 2000 – 4 p.
21. Engine Type: L70MC Vol. III-Components descriptions. MAN B&W Diesel A/S. Copenhagen, Denmark. – 182 p.
22. New K80ME-C and S80ME-C engines. Diesel Facts. №1, 2006. MAN B&W Diesel A/S. Copenhagen, Denmark. – P. 7.
23. Engine operating instructions. Type M32C. Caterpillar Motoren GmbH & Co. KG. 2007. – 2632 p.
24. Engine Type: 6S50MC-C. Index for Description Manual. Vol.03 Description Manual. 902 Piston with Rod and Stuffing Box. Doosan Engine Co. Ltd. Gyeongnam, Korea. – 4 p.
25. The Diesel Engine 9K98ME-C. Instructions Hyundai-MAN B&W Diesel Engines Components No. 1 (Code Book). Hyundai Heavy Industries CO., LTD. Ulsan, Korea. – 157 p.

For preliminary review

26. Sulzer RTA58T. Operating Manual «Marine». Wärtsilä NSD Switzerland Ltd. Winterthur, Switzerland. 1998 – 999 p.
27. Sakabe H. The latest technology of the UE engine with environmentally friendliness, enhanced reliability and high performance. Proceedings of the 7th International Symposium on Marine Engineering Tokyo, October 24-28th, 2005 – 7 p.
28. Service Letter. Cylinder Condition MC Engines. SL00-384/JWF. MAN B&W Diesel A/S. Copenhagen, Denmark. August 2000 – 8 p.
29. Instruction Book L23/30H. Zhenjiang Marine Diesel Works. Guangtangqiao Zhenjiang, China.1996. – 617 p.
30. Engine type ZAV40S. Spare parts catalogue. Wärtsilä Corporation. 2004 – 325 p.
31. Bilousov I., Bulgakov M., Savchuk V. Modern Marine Internal Combustion Engines. Springer International Publishing. 2020. – 385 p.
32. Engine operating instructions. Type M43C. Caterpillar Motoren GmbH & Co. KG. 2004. – 1771 p.
33. Brown D.T., Eckert B.O. The Rotating Piston More Than 50 Years On. SenW 57STE AARGANG NR 6. – P. 329-333
34. 710G Series turbocharged power. Products engines. Electro-Motive Diesel, Inc. Illinois, USA. 2006 – 542 p.
35. MAN B&W Low Speed Large Bore Engines Now with Higher Power Concentration. MAN Diesel, Copenhagen, Denmark. – 16 p.
36. Nerenst P. Service Experience MC Engines. Presented by MAN B&W Diesel Evening, May 15, 2003, Athens. – 195 p.
37. Service Experience. Two-stroke Engines. MAN Diesel & Turbo. Copenhagen, Denmark. 2012. – 32 p.
38. Parts Manual 3516C Marine Auxiliary JTS1-Up (Engine). Caterpillar. 2010 – 723 p.
39. L27/38 Project guide. Diesel & TurboHoleby Generatinc Sets. 1996 – 95 p.
40. Spare Parts Catalogue VASA 32. Wärtsilä Finland Oy, Service. Vaasa, Finland. 2006 – 352 p.
41. https://ru.pinterest.com/pin/637189047264962168/ (19.02.2024)
42. https://asa-equipments.com/ (19.02.2024)
43. Maintenance instruction. Engine type W46. Assessment of bearings. Document No. 4610Q002. 2005 – p.
44. Wärtsilä RT-flex60-C. Spare Parts Code Book «Marine». Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2005 – 332 p.
45. 6L21/31 Index for Maintenance Manual. Section 510. Crankshaft and Main Bearings. HSD – 14 p.
45. Wärtsilä 46. Spare Part Catalogue. Wärtsilä Corporation. – 440 p
46. Hupli J., Troberg M. The smokeless Sulzer ZA40S. Technology, Wärtsilä Italia S.p.A. Marine News – №1 2005 – P. 23-25
47. Workshop Manual TBD 645. Motoren-Werke Mannheim AG. Mannheim. – 160 p.
48. L42MC Mk 6 Project Guide. Two-stroke Engines. 5th Edition., MAN B&W Diesel A/S. 1999 – 240 p.
49. MAN B&W – S60ME-C Vol. III Components Description. Original instructions DS 4207602-1. MAN Diesel & Turbo. – 251 p.
50. Damper. Instruction Handbook. Couplings And Dampers. Geislinger GmbH. Salzburg, Austria – 34 p.

For preliminary review

SECTION 4

Air-gas systems of marine engines

Marine internal combustion engines, like most other piston and rotary vehicles for transport purposes, use air oxygen, which enters the engine from the surrounding atmosphere, to burn fuel in their working space. Interacting with the fuel, a mixture of unreacted air components with combustion gases is formed in the engine working space, which is used as a working fluid that performs useful work during the expansion process. In addition, atmospheric air is used to clean the working cylinders from combustion products. Effective cleaning of the working space with atmospheric air is especially important in two-stroke engines that do not have a displacement piston stroke.

An equally important process is the removal of engine exhaust gases into the environment, the efficiency of which largely determines the economic, environmental and a number of other important indicators of engines.

4.1 Use of atmospheric air in the operating process of marine engines

4.1.1 Atmospheric air is the most important component necessary for organizing the operating process in engines with an open thermodynamic cycle. It is used to form the working fluid at the compression and expansion stage, and at the combustion stage, the oxygen contained in the air is used as an oxidizer.

In addition to the direct use of atmospheric air as the main component for the formation of working fluids, the earth's atmosphere is used by marine engines as a natural refrigerator, into which a significant part of the process heat and fuel combustion products are removed.

Atmospheric air is a mixture of gases, consisting mainly of nitrogen and oxygen. Taken together, the share of these two components is 98...99% depending on the concentration of water vapor (humidity). In addition to the two named components, dry atmospheric air contains in small proportions: carbon dioxide, hydrogen, methane, inert gases and other elements (Table 4.1).

Table 4.1 – Chemical composition of dry air

Substance	Designation	By volume, %	By weight, %
Nitrogen	N_2	78.084	75.5
Oxygen	O_2	20.946	23.15
Carbon dioxide	CO_2	0.03	0.046
Hydrogen	H_2	0.0005	0.00008
Methane	CH_4	0.0002	0.000084
Noble gases	–	0.9365	1.2965
Other components	–	–	0.007323

A significant influence on the concentration of gases in the air is exerted by water vapor, the content of which depends on temperature, humidity, time of year, climate, etc. Thus, at a temperature, 0°C cubic meter of air can contain up to 5 gram water, and at a temperature + 10°C bout 10 gram.

When taken from the atmosphere, air is subjected to filtration, removal of excess moisture and dilution with crankcase gases or combustion products, for example, during exhaust gas recirculation in order to reduce oxygen concentration.

When calculating the engine operating process, the composition of the air entering it can be adjusted depending on the methods used to prepare the charge for supply to the cylinder, but by default it is generally accepted that the main components of the charge are nitrogen and oxygen. The influence of other components of the gas mixture on the operating process can be neglected due to

For preliminary review

their low concentration.

The main stage of the operating process, which actually determines the nature of its flow, is the process of fuel combustion, accompanied by the release of heat. During the oxidation reactions of fuel hydrocarbons with atmospheric oxygen, thermal energy is released and combustion products are formed, which, in a mixture with unreacted components of the gas mixture, form the working fluid in the expansion section.

The combustion process is a set of sequential and parallel chemical reactions, of which, when calculating the combustion process, only the reactions of the formation of final oxidation products of the main combustible fuel components – carbon (C), hydrogen (H) and sulfur (S) are taken into account.

4.1.2 Formation of the charge of the working cylinder and parameters of its state at the moment of compression. To organize an efficient operating process in the engine, an important point is to determine the amount of air required for complete combustion of the fuel.

Theoretically, the required amount of air for 1 kg fuel combustion L_0 is determined by the stoichiometric ratio, that is, the ratio of the combustible components of the fuel with air oxygen in the mixture at which all reagents are consumed during the combustion process. Such a mixture, called stoichiometric, after completion of the combustion process will consist only of the final products of combustion, such as CO_2, H_2O and SO_2 and unreacted nitrogen N_2. The combustion process is determined based on 1 kg the fuel burned, while the amount of air and the resulting gaseous products of complete combustion is measured in kilomoles (kmol is the amount of gas whose mass in kg is equal to its molecular weight).

Total amount of oxygen required for complete combustion C, H and S are determined by summing the amounts of oxygen required for the combustion of each of the combustible elements according to the final reactions:

for carbon $O_2 \rightarrow CO_2$;

$12 + 32 \rightarrow 44$;

for hydrogen $H_2 + ½\ O_2 \rightarrow H_2O$;

$2 + 16 \rightarrow 18$;

for sulfur $S + O_2 \rightarrow SO_2$;

$32 + 32 \rightarrow 64$.

The amount of oxygen required to form a stoichiometric mixture based on 1 kg fuel, in kilomoles, will be:

$$O_{st} = \frac{C}{12} + \frac{H}{4} + \frac{S}{32} - \frac{O}{32},$$

Where C, H, S, O – mass fraction of each component in 1 kg the fuel.

This equation takes into account the oxygen included in the organic structure of the fuel. Its presence in the fuel reduces the thermal effect of combustion reactions, so it is taken into account in the equation with a negative sign.

If we assume that the air contains approximately 23% oxygen O_2 by mass (Table 4.1), the theoretically required number of kilomoles of dry atmospheric air to burn one kilogram of fuel will be under standard conditions:

$$L_0 = \frac{1}{0{,}23}\left(\frac{C}{12} + \frac{H}{4} + \frac{S}{32} - \frac{O}{32}\right).$$

Relationship between molar L_0 (kmol/kg) and mass l_0 (kg/kg) expressions of air theoretically required for combustion is established by the expression:

$$l_0 = \mu_a L_0 = 28.97\ L_0 \approx 14.3 \ldots 14.6\ \text{kg/kg},$$

where $\mu_a = 28.92$ is the mass of one kilomole of air.

In practice, for efficient combustion of fuel in a stoichiometric mixture, a high degree of homog-

For preliminary review

enization is required (uniform distribution of fuel molecules in the charge volume, when the reacting substances are in the same state of aggregation, namely gaseous). If this condition is not met, local zones are formed inside the charge, in which all the oxygen is consumed and further combustion is impossible, and the supply of oxygen from other zones, where it is in excess, physically takes up the time allotted for the combustion process. A high degree of homogenization is achieved in engines with external mixture formation, operating on light liquid (gasoline) or gaseous fuels. In such engines, the fuel to air ratio is maintained close to stoichiometric.

In diesel engines, obtaining homogeneous mixtures is difficult due to the peculiarities of mixture formation, in which fuel enters the combustion chamber in liquid form, forming a heterogeneous system at the first stage with the cylinder charge (when the reactants are in different phase states). During the process of heating and evaporation of the fuel, the system goes from a heterogeneous state to a homogeneous state, but this requires additional time. Limiting the time allotted for combustion significantly affects the completeness of fuel burnout and the efficiency of its use.

Excess air coefficient. To increase the reaction rates during internal mixture formation in diesel engines, an excess of oxidizer is provided in the volume of the combustion chamber, that is, the actual amount of supplied air L significantly exceeds the theoretically required L_0. The relationship between the actual and theoretically required amount of air for combustion is called the *excess air coefficient* and is denoted by – α,

$$\alpha = L/L_0.$$

For a given excess air ratio, the actual number of kilomoles for 1 kg fuel combustion will be:

$$L = \alpha L_0, \text{ kmol/kg}.$$

Air mass:

$$l = 28.95\ \alpha L_0, \text{ kg/kg}.$$

According to experimental data, for modern supercharged diesel engines α varies within the following limits (Table 4.2):

Table 4.2 – Excess air coefficient for various types of marine diesel engines

Low-speed diesel engines	Medium-speed diesels	High-speed diesels
2.5…3.5	2.0…3.2	1.8…2.6

The tendency to increase the excess air ratio in modern marine diesel engines is also due to the need to reduce the maximum cycle temperature, which leads to a decrease in the content of nitrogen oxides in the exhaust gases. Reducing the temperature is especially important for two-stroke engines, where the amount of heat transferred to the piston from the combustion products, other things being equal, is twice as much as in four-stroke engines, which increases their thermal intensity.

However, not all of the air supplied to the cylinders is directly involved in the operating process. Some of it, mixed with exhaust gases, goes into the exhaust system. Thus, the mass of air involved in combustion G_{air} will always be less than the mass of air supplied through the gas distribution elements G_s. This makes it possible to improve the cleaning of the cylinders from combustion products remaining from the previous cycle and ensure optimal temperature conditions for the piston group parts due to additional heat removal.

The ratio $\varphi_a = G_s / G_{air}$, characterizing the air consumption for purging and filling the cylinder and is called the *purge coefficient*. The higher its value, the greater the air consumption for gas exchange and the energy consumption for driving the charging units. In two-stroke low-speed diesel engines that do not have pump strokes φ_a =1.6...1.65, in four-stroke medium- and high-speed diesel engines – φ_a =1.0...1.2.

From the air receiver, a certain amount of air enters the cylinder through the intake valves (in four-stroke engines) or ports (in two-stroke engines). Part of the air (excess purge air) leaves the cylinder along with the combustion products. The rest, most of the air G_{air}, called *mass charge*, remains in the working cylinder at the end of the gas exchange.

For preliminary review

During the gas exchange process, most of the charge spent in the previous cycle, containing combustion products, is removed from the cylinder, and a small part, called *residual gases*, due to the imperfection of the gas exchange process, remains in the cylinder, mixing with the air entering it. In combined engines, atmospheric air with parameters p_0, T_0 enters the compressor of the gas turbine supercharger in which it is compressed to parameters p_k, T_k. Next, the air enters the cooler, where, due to cooling and hydraulic resistance of the cooler, its parameters are reduced to the values p_s, T_s. Next, the air enters the air receiver, from where it enters the working cylinder through the intake parts. In this case, due to losses in the intake tract and valves (Δp_a), the pressure p_s decreases to p_a and the temperature rises to T_a due to heating from the hot walls of the channels and from mixing with residual gases.

By the end of the filling process, the contents of the cylinder G_a will consist of a fresh charge G_{air} and residual gases G_r, i.e. $G_a = G_{air} + G_r$.

Theoretically, at air parameters p_s and T_s in front of the intake elements, the volume of the working cylinder V_s could accommodate the amount of air G_s. The actual amount of air G_a that entered the cylinder at the beginning of the compression process is always less than the theoretical one, which is due to the action of a number of factors:

- aerodynamic losses in the intake tract;
- heating of air by the amount ΔT from contact with the walls of the intake tract and gas distribution elements;
- the presence in the cylinder of residual gases with a temperature T_r and heating from them of incoming air having a lower temperature T_s.

To quantify the perfection of the filling process, the concepts of *residual gas coefficient* γ_r and *filling coefficient* η_f are introduced.

The residual gas coefficient γ_r is the ratio of the amount of residual gases M_r (G_r) to the amount of air L_{air} (G_{air}) entering the cylinder,

$$\gamma_r = M_r / L_{air} \text{ or } \gamma_r = G_r / G_{air},$$

where M_r and L_{air} are expressed in kilomoles per 1 kg of fuel; G_r and G_{air} are expressed in kg.

The coefficient γ_r depends on the stroke, engine speed, the implemented gas exchange method and the design of the intake tract, boost pressure, cylinder dimensions and combustion chamber shape, compression ratio, etc.

In four-stroke engines, an increase in the degree of boost makes it possible to increase the valve overlap angle, which leads to a decrease in γ_r due to better purging of the combustion chamber volume. A decrease in the volume of the combustion chamber V_c with an increase in the compression ratio ε also leads to a decrease in γ_r. Approximate values of γ_r for different types of engines are:

- low speed engines 0.02…0.08;
- medium speed engines 0.01…0.02;
- high-speed engines 0.04…0.12.

Approximately, taking into account a number of assumptions, γ_r can be estimated using the formula:

$$\gamma_r = \frac{T_s + \Delta T}{T_r} \frac{p_r}{\varepsilon p_a - p_r}.$$

The temperature of the residual gases T_r, for low-speed engines lies in the range of 700...800 K, and for medium- and high-speed engines 700...1000 K.

The increase in air temperature from the surfaces of the intake tract ΔT depends on the degree of boost and rotation speed. As the degree of boost increases, the temperature difference between the incoming air and the surfaces decreases, and as the rotation speed increases, the contact time between the air and the walls decreases. For low-speed engines ΔT lies within 5...10 K, and for medium- and high-speed engines 5...20 K.

The filling coefficient η_f is the ratio of the actual amount of air entering the working cylinder G_{air} to the amount G_s that could fill the volume of the cylinder at pressure p_s and temperature T_s of air directly in front of the engine intake parts:

For preliminary review

$$\eta_f = G_{air} / G_s;\ G_s = V_s \rho_s;\ \rho_s = \frac{p_s}{RT_s},$$

where ρ_s is the air density after the compressor.

The filling factor η_f shows what proportion of the theoretically possible charge is the actual charge of the cylinder. It depends on the pressure and temperature of the mixture at the beginning of compression (T_a, p_a), residual gas temperature T_r and residual gas coefficient γ_r. To estimate the value of the filling coefficient η_f from the listed factors, a number of assumptions are made:

- the filling process ends at BDC (point *a*);
- the heat capacities of air $\overline{c}_v'$ and residual gases $\overline{c}_v''$ at the temperature of the mixture at the beginning of compression are the same ($\overline{c}_v' = \overline{c}_v'' = \overline{c}_v'''$).

From the material balance equation at the end of the filling process, you can find the number of moles M_a of the working mixture at point *a,* consisting of incoming air L_{air} and residual gases M_r:

$$M_a = L_{air} + M_r = L_{air}(1 + \gamma_r) \tag{3.13}$$

The values of M_a and L_{air} can be expressed in terms of the corresponding state parameters according to the gas equation of state:

$$M_a = \frac{p_a V_a}{8{,}314 T_a};\ L_{air} = \frac{p_s V_{air}}{8{,}314 T_s} = \frac{p_s V_s \eta_f}{8{,}314 T_s} \tag{3.14}$$

Here it is taken into account that $V_{air} = V_s \eta_f$.

Substituting expression (3.14) into equation (3.13), we obtain:

$$\frac{p_a V_a}{T_a} = \frac{p_s V_s \eta_f}{T_s}(1 + \gamma_r),$$

where:

$$\eta_f = \frac{V_a}{V_s}\frac{p_a}{p_s}\frac{T_s}{T_a}\frac{1}{1+\gamma_r}.$$

By introducing the relation $V_a = \varepsilon V_s$ and $V_s = (\varepsilon - 1)V_s$, we obtain a general formula for determining the filling factor for both four-stroke and two-stroke engines:

$$\eta_f = \frac{\varepsilon}{\varepsilon - 1}\frac{p_a}{p_s}\frac{T_s}{T_a}\frac{1}{1+\gamma_r}. \tag{3.15}$$

When calculating the operating processes of four-stroke engines, the geometric (or nominal) compression ratio is used, since during the delay in closing the intake valves, the pressure in the cylinder changes slightly. In two-stroke engines, the actual compression ratio is usually used, with a fraction of the piston stroke ψ_s *taken for ports or valves that close* last during compression.

The filling coefficient of two-stroke engines, calculated using formula (3.15), refers to the useful part of the piston stroke:

$$\eta_f = \frac{\varepsilon_0}{\varepsilon_0 - 1}\frac{p_a}{p_s}\frac{T_s}{T_a}\frac{1}{1+\gamma_r}(1 - \psi_s), \tag{3.15a}$$

where ε_0 is the actual compression ratio.

From formulas (3.15) and (3.15 *a*) it follows that with a decrease in the resistance of the intake tract and an increase in r_a, as well as with a decrease in the charge temperature at the end of filling the filling factor will increase. For two-stroke low-speed engines, the filling factor η_f is in the range of 0.75...0.85, for medium- and high-speed diesel engines it is 0.85...0.95.

The main parameters of the filling process are pressure p_a and temperature T_a of the working mixture before compression begins. These parameters depend on the pressure p_s and temperature T_s in the air receiver. The pressure at the end of filling the cylinder can be estimated:

For preliminary review

- for low-speed two-stroke engines $p_a \approx (0.85\ldots0.95)\, p_s$;
- for medium- and high-speed four-stroke engines $p_a \approx (0.96...0.99)\, p_s$.

In turn, p_s depends on the pressure value at the outlet of the supercharger compressor p_k, which is associated with a given value of the average effective pressure p_e:

- for low-speed two-stroke engines $p_k \approx (0.18\ldots0.22)\, p_e$;
- for medium- and high-speed four-stroke engines $p_k \approx (0.16...0.21)\, p_e$.

Losses in the air cooler are usually $\Delta p \approx 0{,}002...0{,}006$ MPa.

The air temperature in the cylinder at the end of filling T_a depends on T_0 and the temperature after the cooler T_s, in which the purge air is cooled to a temperature T_s. As a result of air cooling, its density ρ_s and, consequently, its mass charge increase. The most optimal temperature for these operating conditions is considered to be 313…323 K (40…50°C), since at lower values there is an intense release of moisture from the air, which can worsen the operating conditions of the elements of the cylinder-piston group.

During the filling process, air from the receiver, having a temperature T_s, is heated from the walls and channels of the cylinder to $T'_s = T_s + \Delta T$ and is additionally heated when mixed with residual gases having $T_r = 700\ldots1000$ K.

Thus, at the end of filling, a gas mixture with a temperature of T_a, which depends on the temperature of the incoming air T_s, in the amount of L_{air} with heat capacity $\overline{c}'_v$ heated on the hot walls of the cylinder to a temperature $T_a = T'_s = T_s + \Delta T$ and residual gases in the amount of M_r with temperature T_r and heat capacity $\overline{c}''_v$.

The heat balance equation for the mixture at the end of filling has the form:

$$L_{air}\overline{c}'_v T'_s + M_r \overline{c}''_v T_r = \left(L_{air} + M_r\right) \overline{c}'''_v T_a, \text{ or}$$

$$\overline{c}'_v G_{air} T_{air} + \overline{c}''_v G_r T_r = \overline{c}'''_v G_a T_a$$

where $\overline{c}'''_v$ is the heat capacity of the gas mixture.

Taking the assumption of equality of heat capacities $\overline{c}'_v = \overline{c}''_v = \overline{c}'''_v$ (air, residual gases and mixture) and dividing both sides of the equation by L_{air}, taking into account that $M_r/L_{air} = \gamma_r$, we obtain:

$$T'_s + \Delta T + \gamma_r T_r = \left(1 + \gamma_r\right) T_a,$$

where

$$T_a = \frac{T_s + \Delta T + \gamma_r T_r}{1 + \gamma_r}. \qquad (3.16)$$

From (3.16) it is clear that the temperature of the working mixture T_a increases with increasing coefficient of residual gases γ_r and the degree of heating ΔT and depends relatively little on temperature T_r.

4.2 Composition of combustion products and exhaust gas parameters.

4.2.1 The composition of the gas mixture formed in the cylinder at the end of the combustion process, in addition to the combustion products CO_2, H_2O and SO_2, includes oxygen O_2 (due to its excess) and nitrogen N_2, which does not participate in combustion (nitrogen compounds are also found in fuel), as well as products of incomplete combustion (CO, CH, Particulate matter (soot – C)) and *products of oxidation of air nitrogen* (NO_x).

According to the MAN company, a modern 18V48/60 medium-speed engine with an optimized operating process for nitrogen oxide emissions (NO_x), running on heavy fuel with a sulfur content of 4%, emits up to 136 t/h of exhaust gases into the atmosphere, of which 460 kg about harmful substances. This is approximately 0.35% of the total mass of gases, including NO_x 0.17%, sulfur dioxide (SO_2) 0.15%, hydrocarbons (CH) 0.02%, carbon monoxide (CO) 0.007% and particulate matter (PM) 0.003% (Fig. 4.1).

For preliminary review

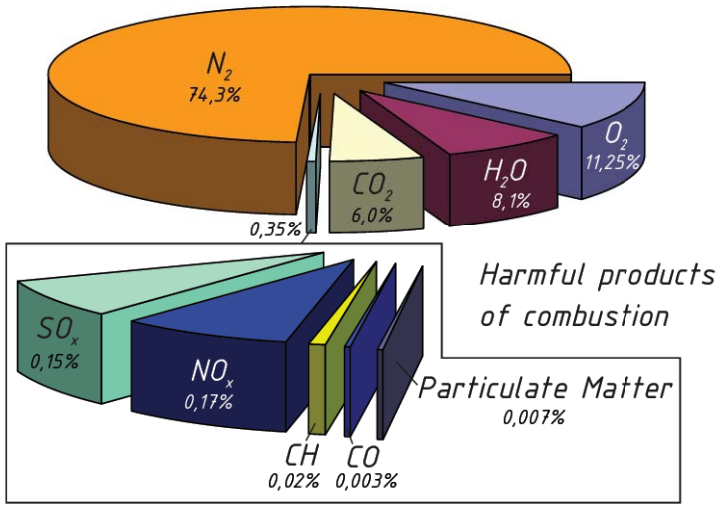


Figure 4.1 – Approximate composition of combustion products of a marine engine when using heavy fuel containing 4% sulfur (adapted from [1])

The amount of *complete combustion products* formed during 1 kg fuel combustion is determined by the previously given stoichiometric combustion equations (in kmol/kg of fuel):

$$M_{CO_2} = \frac{C}{12};\ M_{H_2O} = \frac{H}{2};\ M_{SO_2} = \frac{S}{32}.$$

During combustion with excess air ($\alpha > 1$), the oxygen entering it does not participate in the reaction in full. Amount of excess oxygen O_2:

$$L_{O_2} = \alpha L_0 - L_0 = (\alpha - 1)L_0,\ \text{kmol/kg fuel},$$

hence:

$$M_{O_2} = 0{,}21(\alpha - 1)L_o,\ \text{kmol/kg fuel}.$$

The remaining number of moles in the combustion products comes from inert nitrogen N_2, which is part of the air:

$$M_{N_2} = 0{,}79L_{O_2} = 0{,}79\alpha L_0.$$

The total quantity M of products of complete combustion of one kilogram of fuel at $\alpha > 1$ is determined by summing all components:

$$M = \frac{C}{12} + \frac{H}{2} + \frac{S}{32} + (\alpha - 0{,}21)L_0,\ \text{kmol/kg fuel}.$$

From the law of conservation of mass, it follows that the mass of combustion products is equal to the sum of the masses of air and fuel before combustion, while the volumetric quantities of the working fluid before and after combustion are not equal. During chemical combustion reactions, the number of molecules of gaseous products increases, resulting in an increase in volume (number of kilomoles). This increase can be determined by the difference:

$$\Delta M = M - L = M - \alpha L_0.$$

To estimate the increase in volume (number of moles) of combustion products, *the theoretical (chemical) coefficient of molecular change is used* β_0, which represents the ratio of the number of moles of combustion gases M to the number of moles of air L without taking into account the residual gases in the cylinder:

$$\beta_0 = \frac{M}{L} = \frac{L + \Delta M}{L} = 1 + \frac{\Delta M}{L}.$$

In addition to the components listed above, the combustion products contain residual gases M_r,

For preliminary review

which remain in the cylinder from the previous cycle and also affect the volume of combustion products.

Attitude

$$\gamma_r = M_r / L,$$

those. the amount of residual gases to the actual amount of fresh charge is called *the residual gas ratio.*

Taking this into account, the concept *of the calculated (real) coefficient of molecular change* β is introduced, which is the ratio of the number of moles of gas after combustion to the number of moles of air, taking into account the presence of residual gases, i.e.

$$\beta = \frac{M + M_r}{L + M_r} = \frac{\beta_0 + \gamma_r}{1 + \gamma_r}.$$

The value of the calculated coefficient of molecular change for hydrocarbon fuels lies in the range β = 1.03...1.04.

Important cycle parameters for supercharged engines are the pressure p_g and temperature T_g of the exhaust gases in the exhaust pipeline, since the potential that can be activated in the gas turbine and/or in the recovery circuit of the ship's power plant depends on them.

The average temperature of combustion products can be determined from the polytropic equation based on the gas temperature at the end of expansion T_b:

$$T_r = \frac{T_b}{(p_b / p_r)^{\frac{n_2 - 1}{n_2}}}, \qquad (3.36)$$

where n_2 is the expansion polytrope index; p_r – exhaust gas pressure at the end of expansion.

Having accepted the assumptions that:

- the process of gas outflow occurs in the absence of heat exchange with the walls;
- the speed of gases upon expiration drops to zero;
- residual gases do not participate in the process under consideration.

You can write down the heat balance equation when mixing products in the outlet receiver:

$$\overline{c}_p'' \, M T_r + \overline{c}_p' L(\varphi_a - 1) \, T_s = \overline{c}_p''' \left[M + L(\varphi_a - 1) \right] T_g, \qquad (3.37)$$

High values of φ, characteristic of two-stroke engines, lead to a decrease in the temperature of the exhaust gases, which mix in the exhaust receiver with excess scavenging air, which has a lower temperature.

Dividing all terms of equation (3.37) by L, after transformation we obtain an expression for determining the temperature of the exhaust gases:

$$T_g = \frac{\overline{c}_p'' \, \beta_0 T_r + \overline{c}_p' (\varphi_a - 1) \, T_s}{\overline{c}_p''' (\beta_0 + \varphi_a - 1)}, \qquad (3.38)$$

where $\overline{c}_p'$, $\overline{c}_p''$, $\overline{c}_p'''$ are the average molar isobaric heat capacities of air, combustion products and the gas-air mixture in the exhaust manifold.

The average temperature of the exhaust gases depends on the engine stroke, the temperature of the gases in the cylinder at the beginning of exhaust, the load and engine speed, the excess scavenging air ratio, the completeness of the fuel combustion process and the presence of afterburning, the pressure in the exhaust manifold, heat exchange in the gas path and other factors.

For marine diesel engines, the exhaust gas temperature is usually within the following limits (K):

- low speed engines 520…650;
- medium speed engines 620…750;
- high-speed engines 720…900.

For preliminary review

4.2.2 Harmful components in combustion products

All combustion products are divided into conditionally neutral and harmful. Neutral products include such products of complete combustion as water vapor (H_2O), carbon dioxide (CO_2), although the latter is classified as a so-called «greenhouse gas» with which global warming is associated. Harmful products include sulfur combustion products (SO_x) and incomplete combustion products, such as carbon monoxide (CO), hydrocarbons (CH), *particulate matter* (PM), as well as nitrogen oxidation products (NO_x).

Sulfur oxides, designated SO_x, include sulfur monoxide (SO), sulfur dioxide (SO_2) and sulfur trioxide (SO_3). Once in the atmosphere, sulfur monoxide SO is immediately oxidized to SO_2, which is a gas that does not decompose under atmospheric conditions. Sulfur trioxide SO_3 enters the atmosphere as PM. Sulfur oxidation products are air pollutants along with NO_x, which lead to acid rain. It is for this reason that MARPOL limits on SO_x emissions are becoming more stringent every year.

The International Maritime Organization (IMO) establishes emission control areas, which impose stricter controls to minimize emissions of sulfur compounds into the atmosphere (SECA). The amount of SO_x emissions for marine engines is regulated by Annex VI to MARPOL, namely paragraph A.2 of Regulation 14. Instead of directly regulating the amount of SO_x in emissions, a limit is set on the permissible sulfur content in the fuel. From January 1st 2020. By amending MARPOL Rule 14, the International Maritime Organization has limited the permissible amount of sulfur and its compounds in fuels to 0.1% within the emission control areas (ECA) and SECA zones and 0.5% in all other zones. The use of fuels with sulfur content higher than 0.5% is permitted only on ships equipped with exhaust gas scrubbing systems. Owners and operators of all other vessels are required to use only low-sulfur marine fuels or their alternatives. Violation of these rules is fraught with heavy fines, detention of ships, and in some jurisdictions even imprisonment.

Carbon monoxide (CO) in diesel engines is formed during cold-flame reactions during combustion in conditions of lack of oxygen and also due to the dissociation of carbon dioxide (CO_2) at high temperatures.

Hydrocarbons are a generalized name for more than 200 individual substances formed in the working space of an engine, differing mainly in the number of carbon atoms in the molecule. In this case, the composition of hydrocarbons present in the exhaust gases differs from the initial composition of fuel and oil hydrocarbons, since their thermal decomposition occurs under the influence of high temperature. Hydrocarbon compounds are formed in the exhaust gases due to flame attenuation near the relatively cold walls of the combustion chamber, as well as in the so-called «sandwiched» volumes, for example, in the gap above the upper compression ring between the piston and the cylinder. The reason for the appearance of hydrocarbons in the exhaust gases can be not only incomplete combustion of fuel, but also lubricating oil, as well as the entry of fuel vapor into the cylinder during expansion, for example from the valve sac of the fuel injector.

Currently, neither the IMO nor other international regulatory organizations have established any standards for emissions of carbon monoxide (CO) and hydrocarbons (CH) in combustion products of marine engines. The content of these substances in exhaust gases largely depends on the completeness of fuel combustion, so all manufacturers pay considerable attention to reducing these emissions, based more on economic than environmental considerations. However, these compounds have a significant impact on the human housing, and their significant accumulation in the air can have serious negative consequences.

Solid particles are a group of compounds that are formed during physical and chemical processes in the working space of the engine. The basis of solid particles, as a rule, is soot (C) on the surface of which heavy hydrocarbons, ash particles, some wear products, solid sulfur compounds, etc. can be absorbed. The presence of solid particles in engine exhaust gases leads to a loss of transparency (increase in optical density) and the appearance of black smoke. The optical density of exhaust gases depends on the amount and size of solid particles; when their concentration in the exhaust gases is more than 1.1 g/m^3, diesel smoke becomes visible. Most solid particles have sizes of 0.4...5 μm.

The formation of soot during fuel combustion in diesel engines and its amount in the exhaust

For preliminary review

gases depends on the conditions of mixture formation and combustion, which are determined by such factors as the operating mode of the diesel engine, the chemical and physical properties of the fuel, the design features of the combustion chamber, the parameters of the fuel equipment, etc.

Incomplete combustion contributes to the formation of soot, which in turn, accumulating on the walls of the combustion chamber leads to a decrease in its thermal conductivity, which causes local overheating and a decrease in the service life of the combustion chamber elements, accelerated wear of the piston rings, etc.

Nitrogen oxides. Diatomic molecular nitrogen (N_2) contained in air is a relatively inert gas, but at elevated temperatures (above 2000 °C) its molecule breaks down into atomic nitrogen (N) which can have ionization levels (called valence states) from plus one to plus five. Thus, nitrogen can form several different oxides. In addition to the so-called «thermal» mechanism for the formation of nitrogen oxides, other mechanisms are also distinguished, but their contribution to the total content of these compounds in engine exhaust gases is insignificant.

Nitrogen monoxide (NO) and nitrogen dioxide (NO_2) make up the majority of nitrogen oxides NO_x (90...95%), other compounds are present in exhaust gases in much smaller quantities. In the working cylinders of diesel engines, NO is mainly formed during the operating process. When it enters the atmosphere, it is oxidized to NO_2 and, in much smaller quantities, to N_2O_5. Therefore, the current standards in accordance with the requirements of Annex VI of the MARPOL 73/78 Convention regulate only nitrogen dioxide (NO_2) as a general compound of the entire family, as the most common form of NO_x in the atmosphere generated by anthropogenic (human) activities.

The emission of even small amounts of NO_2 is a serious problem due to the high toxicity of this component of exhaust gases and its active participation in photochemical reactions in the atmosphere. The stages of tightening IMO standards for emissions of nitrogen oxides into the atmosphere are presented in Fig. 4.2.

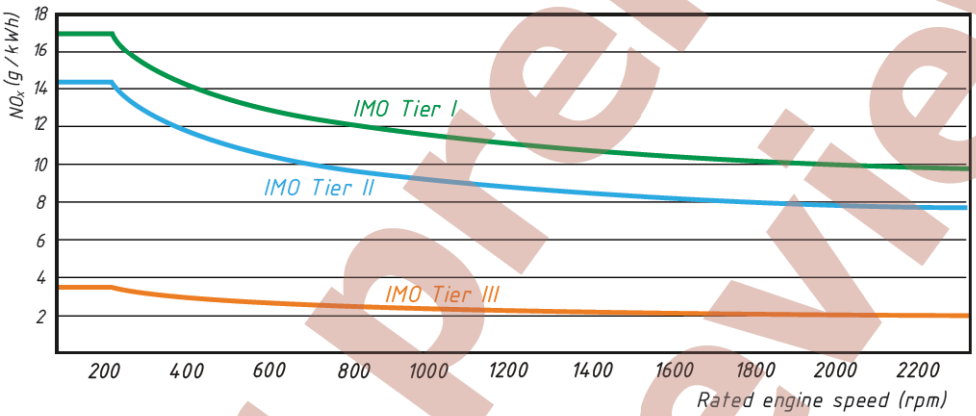


Figure 4.2 – Curves for gradual limitation of NO_x emission standards for marine internal combustion engines: Tier I (2000); Tier II (2011); Tier III (2016) (adapted from [1])

4.3 Gas-dynamic processes during gas exchange

In Fig. 4.3 shows typical curves of pressure changes during the gas exchange process of a four-stroke engine combined with the opening profiles of the gas distribution valves. It can be seen that after opening the exhaust valve, the pressure in the working cylinder decreases sharply, which corresponds to the free exhaust phase under the influence of the pressure difference between the cylinder and the exhaust manifold (section *I*). The throughput of the exhaust valves is selected in such a way that by the time the intake valve opens, the pressure in the working cylinder is equal to or less than the pressure in the intake receiver. Otherwise, combustion products may be thrown from the cylinder into the intake receiver.

The entry of exhaust gases from the working cylinder leads to a pressure surge in the exhaust manifold. As a result, gas-dynamic release processes have a pulsed nature with wave amplitudes up to (1.5...1.6) p_s.

The change in pressure in the exhaust manifold p_{ep} depends on how the further process is organized:

For preliminary review

- release of gases into a common manifold from all cylinders;
- release of gases into a separate pipe.

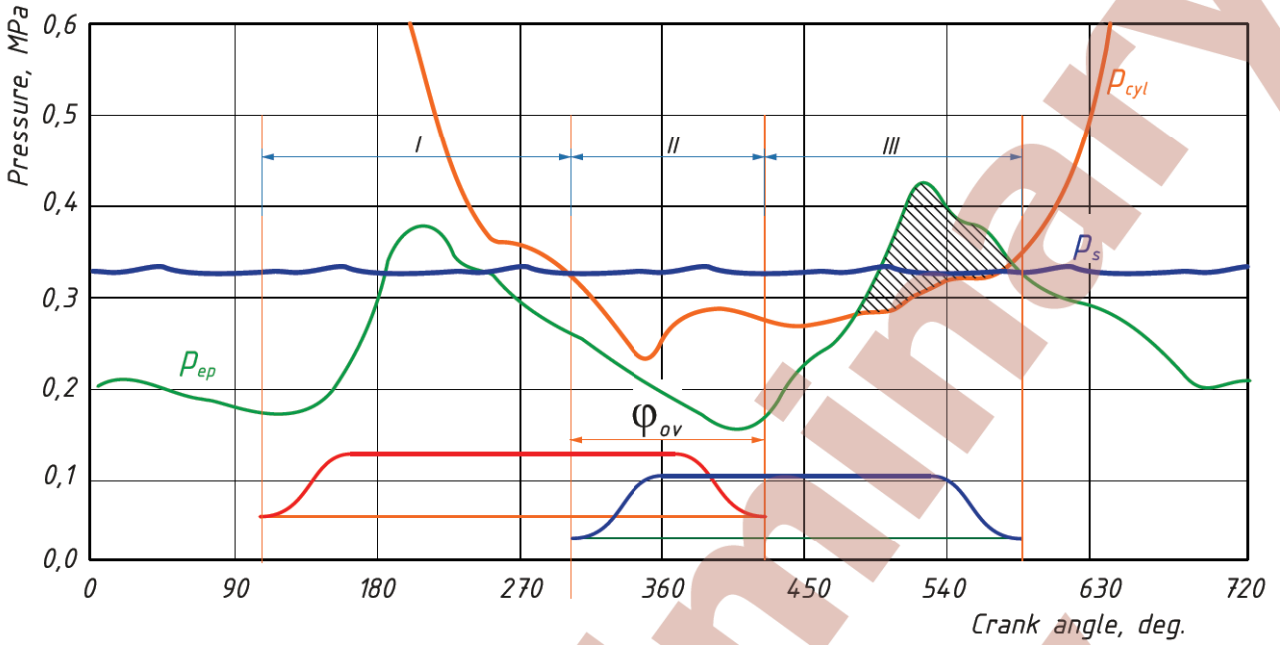


Figure 4.3 – Changes in pressure in the working cylinder p_{cyl}, inlet receiver p_s, exhaust manifold p_{ep} during the gas exchange process of a four-stroke engine combined with the opening profiles of the gas distribution valves (adapted from [2])

For successful purging of the combustion chamber, the valve overlap section (φ_{ov}, section *II*) is selected in such a way that at this interval the pressure in the intake receiver p_s is higher than the pressure in the exhaust manifold p_{ep}. The higher the boost pressure, the longer the valve overlap can be, the better the cleaning of the cylinder from combustion products can be done. Section *III* shows a case where an increase in pressure in the common exhaust manifold associated with exhaust in one cylinder leads to exhaust gases being thrown into the other cylinder (shaded area) through the open intake valve. Thus, fluctuations in pressure p_g (Fig. 4.3) from the exhaust of neighbouring cylinders can create back pressure at the exhaust and make it difficult to clean the cylinder. To avoid this harmful phenomenon, the exhaust from individual cylinders with similar operating orders is carried out into separate exhaust manifold pipes, which are combined into one pipe at a distance from the valve sufficient to reduce the influence of pressure pulses on neighbouring cylinders.

In *two-stroke engines* with direct-flow valve scavenging, the air flow moves along the cylinder axis with layer-by-layer displacement of combustion products through a valve installed in the cylinder cover. Since in this type of engine there are no piston pump strokes, the filling process is combined with the exhaust process. These processes occur only during part of the piston stroke, the magnitude of which is determined by the moment the exhaust valve closes (Fig. 4.4). All modern two-stroke low-speed engines have purge ports located in the lower part of the cylinder liner; the phases and laws of their opening and closing are determined by the kinematics of the piston movement. The opening and closing phases of the exhaust valve, the height of its lowering, and the law of motion are set by the profile of the camshaft cam or by an electronically controlled hydraulic actuator.

The gas exchange process in a two-stroke engine is conventionally divided into four phases (Fig. 4.4 *a*): free exhaust I; forced release II; purging and filling III, charge loss IV. In this case, phases II and III occur simultaneously, fresh air enters the cylinder through purge ports opened by the piston as it moves to BDC, and combustion products displaced by air are removed through an open exhaust valve located in the cylinder cover. The change in the flow sections of the gas exchange organs according to the angle of rotation of the crankshaft is called «area-time» diagrams

For preliminary review

(Fig. 4.4 *a*). When constructing area-time diagrams, the opening value of ports or a slot under the valve is plotted along the ordinate axis, and the opening time or the angle of rotation of the crank is plotted along the abscissa axis. Therefore, the areas limited by the curves (Fig. 4.4 *a*) represent the time multiplied by the cross-section (area-time) of the opening of gas exchange organs in the corresponding periods that determine their throughput.

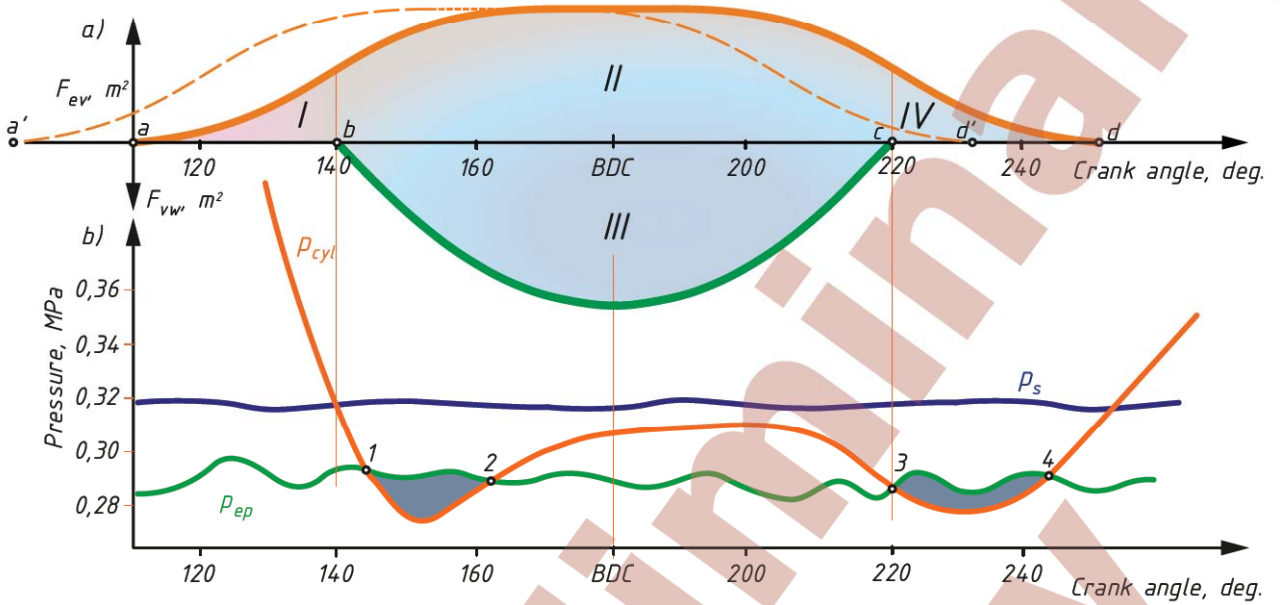


Figure 4.4 – Gas exchange process in a two-stroke engine with direct-flow valve scavenging *a* – diagram of the available «area-time»; *b* – pressure curves

According to the diagram in Fig. 4.4 and the exhaust valve opens first at point *a* – approximately 70° before the piston reaches BDC. The first stage (I) begins – the process of free release. Combustion products rush into the exhaust system at high speed, the pressure in the cylinder p_{cyl} drops sharply (Fig. 4.4 *b*) and by the time the piston, moving down, opens the purge ports (approximately 40° before BDC, point *b* in Fig. 4.4 *a*), it must be less than or equal to the pressure in the purge air receiver p_s to ensure air flow into the cylinder. Otherwise, if by the time the purge ports are opened, $p_{cyl} > p_s$, hot gases may be thrown from the cylinder into the receiver. This can cause a fire in the sub-piston cavity due to the combustion of oil vapors accumulated in the receiver.

Further, as the piston lowers in the cylinder, the purge ports open, increasing the flow area, which reaches a maximum at BDC.

The pressure p_{cyl}, having reached the value p_s, continues to fall and, despite the beginning of air entering the cylinder, decreases even below the pressure in the exhaust manifold p_e (section 1-2, Fig. 4.4 *b*). This is explained by the injection effect of a gas flow moving at high speed in the pipe behind the exhaust valve. The continued flow of air from the receiver leads to an increase in pressure in the cylinder, which approaches p_s. During this period, in section *b-c*, through an open valve, combustion products continue to be released from the cylinder under the influence of air displacing them (periods II and III).

After passing BDC, the piston gradually reduces the flow area of the purge ports until they completely overlap approximately 40° after BDC. We stop filling the cylinder with air, and the pressure in it, due to the same injection action, begins to fall (section 3-4, Fig. 4.4 *b*) below the pressure p_{ep}. Then, as the exhaust valve closes, the cross-section of the gap under it decreases (section *c-d* Fig. 4.4 *a*), the valve begins to throttle the gas flow displaced by the piston. As a result, the pressure in the cylinder begins to increase and by the time the valve closes it is above p_s. The last IV phase of the gas exchange process (section *c-d*, Fig. 4.4 *a*) is accompanied by a loss of charge, which is displaced from the working cylinder into the exhaust receiver through the still open exhaust valve. To reduce charge losses in engines with direct-flow valve scavenging, asymmetrical valve timing is

For preliminary review

used, in which the closing of the exhaust valve occurs as close as possible to the closing of the scavenging ports (dashed curve *a'-d'* in Fig. 4.4 *b*).

The area-time diagram of the purge ports is a combined diagram of their linear dimensions and the piston path curve as a function of the crank angle. For an exhaust valve, the «area-time» is defined as the product of its displacement and the perimeter of the channel opened by the valve. In Fig. 4.5 shows a «area-time» diagram of an engine with a direct-flow gas exchange scheme. To construct a diagram, the scales of the coordinate axes are first adopted: the scale of linear dimensions *H* (ordinate scale) and the scale of rotation angles φ (abscissa scale).

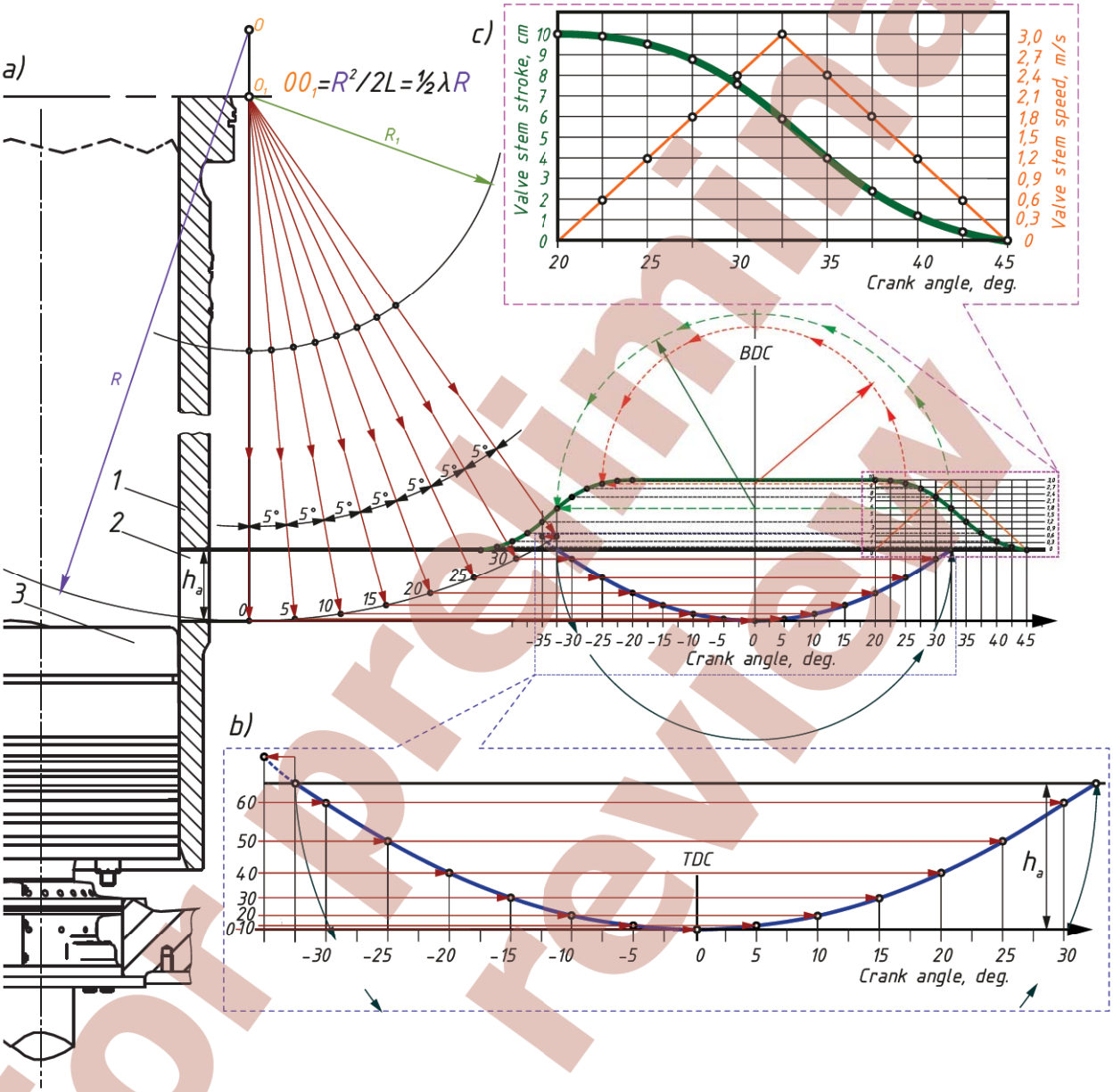


Figure 4.5 – Construction of a «area-time» diagram of the gas exchange organs of a two-stroke engine with direct-flow valve purge: *a* – construction of a bicentric diagram; *b* – construction of a «area-time» diagram of purge ports; *c* – plotting a «area-time» diagram of the exhaust valve

On the accepted scale of linear dimensions *H*, the diagram (Fig. 4.5 *a*) shows a diagram of the purge ports of the cylinder liner and exhaust valves. At the level of the lower edge of the piston when it is in BDC, a horizontal line is drawn, which is taken as the abscissa axis of the area-time diagram. Draw horizontal lines at the level of the top and bottom edges of the ventilation ports. Next, a segment *R* corresponding to the radius of the crank on the accepted scale *H* is plotted from the abscissa axis, and the conditional centre of rotation of the crank is obtained *O* (Fig. 4.5 *a*). To

For preliminary review

take into account the influence of the finite length of the connecting rod on the movement of the piston, a bicentric Brix diagram is constructed for which the second center is O_1 found from the expression $OO_1 = R^2/2L = ½\lambda R$ called the Brix correction, where L is the length of the connecting rod, and λ is the constant of the crank mechanism ($\lambda = R/L$). To the right of it, a curve of the piston path is plotted in a rectangular coordinate system.

Ordinates corresponding to the rotation angles in Fig. 4.5 *b*, represent the linear opening values of the exhaust ports on a scale *H*. The area of the diagram, limited by the piston path curve, the level of the upper edge of the ports and the ordinates corresponding to the angle of rotation of the crank at the beginning and end of the gas exchange period, represents the value of the integral of the linear openings of the ports. Its actual value is determined by multiplying the area by the scale of linear dimensions and the scale of rotation angles.

The kinematics of valve movement depends on the type of its drive mechanism. For systems with a mechanical camshaft drive, a motion law based on a given velocity triangle is often used. This construction principle is shown in Fig. 4.5 *c*. For a given angular gap, the maximum valve opening speed is adopted based on the conditions of reliable operation of the drive mechanism. Next, a triangle of speeds is built. By integrating the velocity curve, the law of valve movement is obtained. Typically, the valve lifting and lowering profiles are made symmetrical, since in reversible engines this makes it possible to ensure the same gas exchange conditions when operating in forward and reverse gear.

In engines with a hydraulic valve drive from electronically controlled actuators, the valve opening phases can be optimized for a specific engine operating mode by having the valve opening and closing angles, the law of its movement and the magnitude of its stroke.

4.4 Purpose and main functions of air-gas systems

Based on the significant role that atmospheric air plays in the operating process of piston engines, as well as the processes of gas exchange between the engine and the environment, it becomes clear how important for any engine is the effective operation of the air-gas system, which is entrusted with the functions of preparing air for supplying it to cylinders and efficient exhaust gas removal.

The air-gas system serves for:

- receiving, cleaning and supplying air to the working cylinders in the amount necessary for efficient combustion of fuel, with a given pressure and temperature;
- reducing intake noise;
- purging the engine working space from combustion products;
- removal of exhaust gases and reduction of exhaust noise;
- recycling the heat of exhaust gases and reducing their toxicity.

The air-gas system, in addition to the engine *gas distribution mechanism*, also includes a number of units and devices interconnected by common mass and energy flows:

- devices for receiving, purifying air and reducing noise at the inlet;
- boost units;
- devices for removal, release and noise reduction of exhaust gases;
- exhaust gas heat recovery units;
- devices and units for reducing exhaust gas toxicity.

The general layout of the air-gas system of a modern LSE is shown in Fig. 4.6.

There is no clear boundary between the functions of individual elements; the same units and components of the air-gas system can perform several functions at once. The presence of certain elements is determined by the type of engine, its design features, as well as the adopted thermal circuit of the entire power plant.

Preparing air for supply to the working cylinder means cleaning it, precompressing it, and regulating temperature and humidity. In some cases, it is envisaged to regulate its chemical composition, for example, by bypassing part of the exhaust gases into the intake tract or by adding combustible

For preliminary review

gases used as fuel to the air.

The efficiency of exhaust gas removal into the atmosphere means their release with minimal back pressure, the use of exhaust gas energy, both to obtain mechanical work and thermal energy, and changing the chemical composition of exhaust gases in order to reduce the content of harmful components in them.

The issue of effective gas exchange between the working cylinders of internal combustion engines and the external environment has become relevant almost from the moment of their creation, and remains so to the present day.

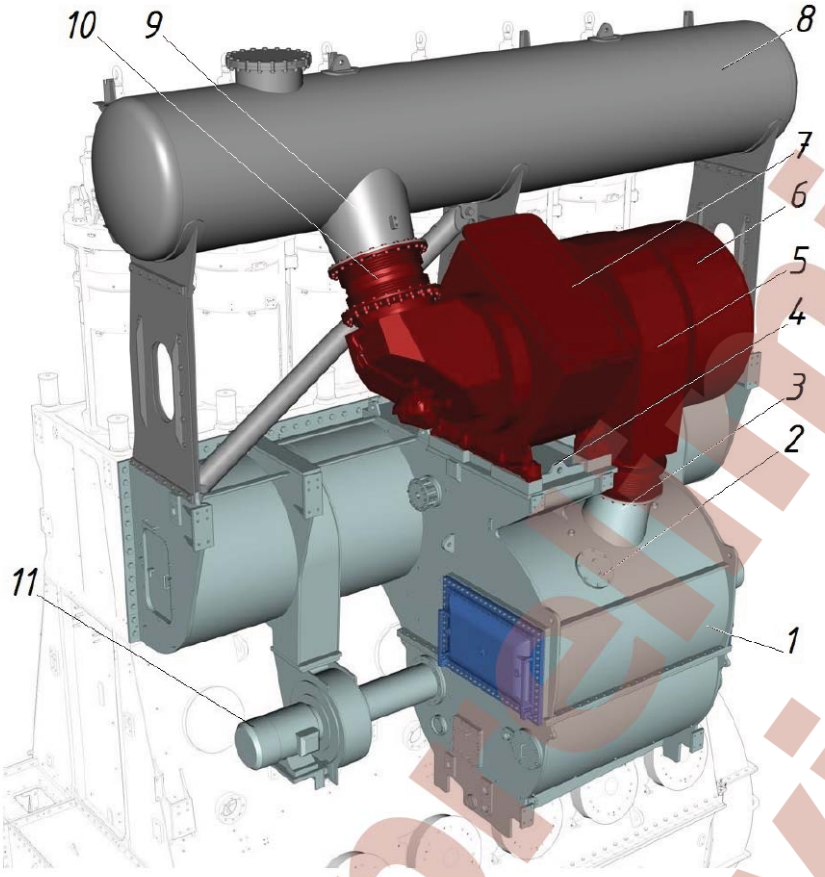


Figure 4.6 – General arrangement of elements of the air-gas system of a modern LSE from MAN: 1 – housing of charge air coolers and moisture separators; 2 – hatch for monitoring and cleaning the cooler; 3 – air duct; 4 – platform for turbochargers; 5 – compressor part of the turbocharger; 6 – air filter; 7 – turbocharger turbine; 8 – exhaust receiver; 9 – gas duct; 10 – thermal expansion compensator; 11 – exhaust valve assembly; 12 – air receiver; 13 – electric blower (adapted from [3])

Over the entire period of existence of piston engines, the mechanisms and assemblies of the air-gas path have gone through a certain development path from low-power engines with cylinder filling under atmospheric pressure to modern systems equipped with supercharging units that make it possible to increase engine power several times.

4.5 Gas distribution mechanisms of marine engines

4.5.1 History of the creation and development of gas distribution mechanisms for marine piston internal combustion engines

When developing the first piston engines, their creators relied heavily on the experience of designing piston steam engines, borrowing from there both ready-made technical solutions and individual components and assemblies. Thus, in the first piston internal combustion engine, which had some commercial success, its creator, the Belgian engineer Etienne Lenoir*, used a spool valve gas distribution system, borrowed from steam engines with only minor modifications (Fig. 4.7).

However, spool-type mechanisms with a sliding spool plate, with the help of which steam was distributed under pressure, turned out to be ineffective in internal combustion engines, in which the working space was filled under atmospheric pressure. The small flow area of the gas distribution organs, large friction losses, which increased significantly with increasing rotation speed, the complexity of the drive from the additional crankshaft journal and a number of other disadvantages

For preliminary review

forced us to very quickly abandon the use of such mechanisms in internal combustion engines.

August Otto* managed to significantly improve the design of the Lenoir engine. In 1863, he created a two-stroke engine with a spool valve. Faced with the same difficulties as Lenoir, Otto made a number of attempts to improve the spool mechanism.

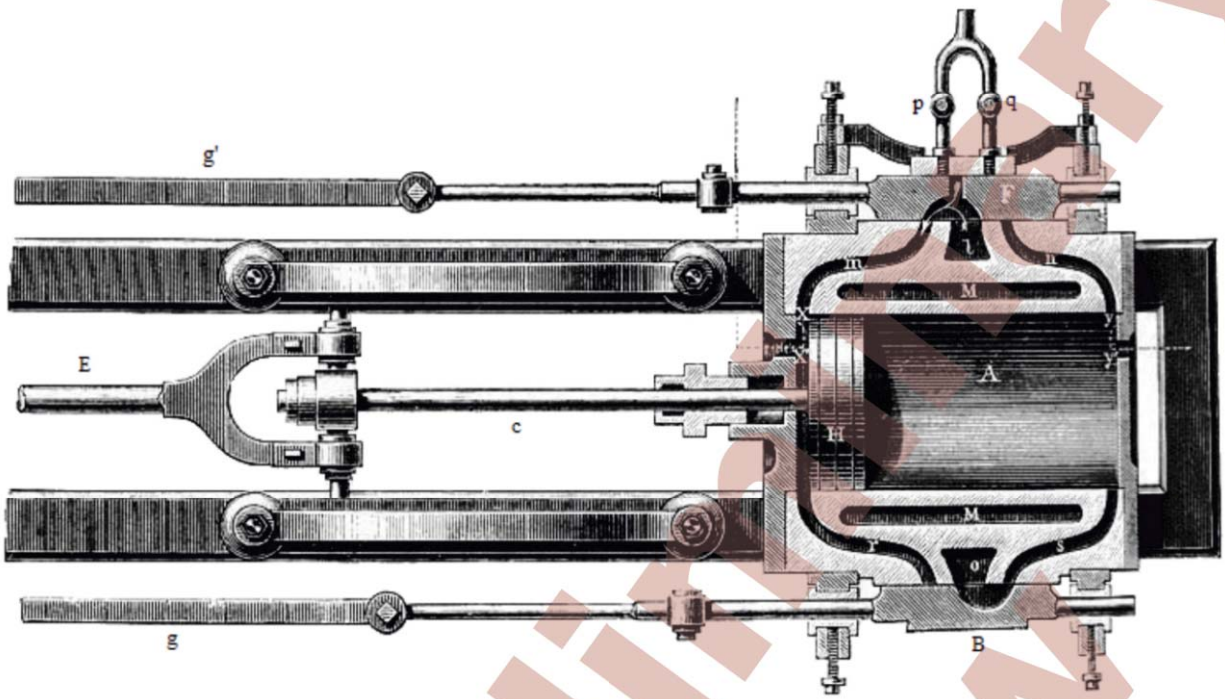


Figure 4.7 – Spool gas distribution mechanism of the Lenoir engine (adapted from [4])

Despite the commercial success of his engine, which he launched together with Karl Eugen Langen, Otto continued his development of a four-stroke engine. This work ended in success in 1876. In the design of a four-stroke engine, the problem of reducing intake resistance was solved by installing an automatic intake valve (pos. *F*, fig. 4.8), which opened under the influence of vacuum in the working cylinder. The exhaust gases were released using a spool pair (pos. *D*, fig. 4.8).

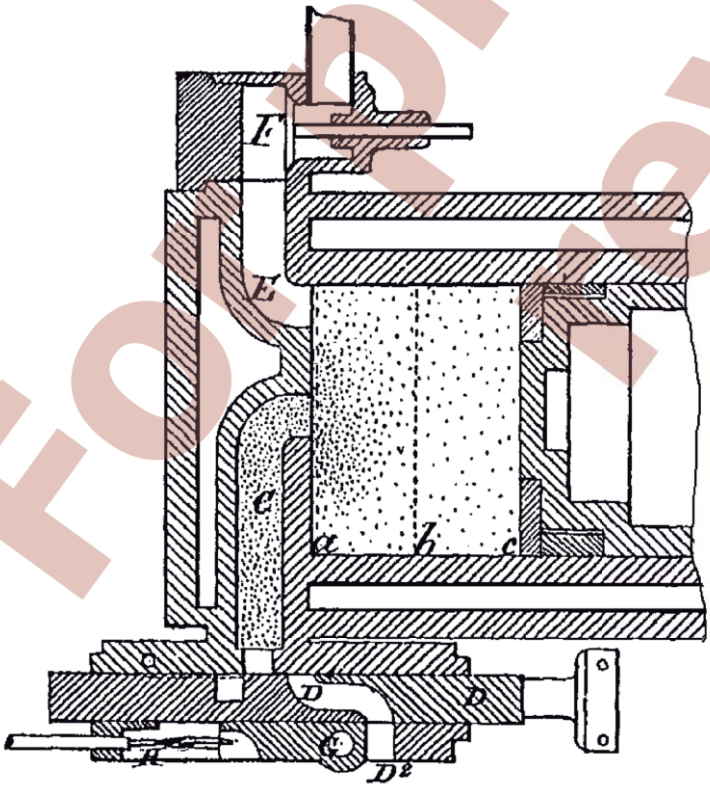


Figure 4.8 – Combined spool-valve gas distribution of a four-stroke Otto engine (adapted from [4])

For preliminary review

It should be noted that at the dawn of the development of the first internal combustion engines, there was no unified approach to the design of gas distribution elements. They tried to apply already known technical solutions to newly created internal combustion engines, and valve timing was one of them. This type of steam distribution has already been tested on some types of steam engines and has proven its effectiveness.

Many engineers and designers of that time came to the idea of using valve mechanisms to organize gas exchange between the working cylinder and the environment, and today it is perhaps not possible to prioritize such a solution for any of them. At the same time, the first engine with a valve-type gas distribution mechanism that had great commercial success can be considered the engine built by Gottlieb Wilhelm Daimler* in 1885 (Fig. 4.9).

In the Daimler engine, as in the Otto engine, the intake valve opened under the influence of vacuum, and the exhaust valve already had a mechanical drive from the engine crankshaft on which there was a system of profiled guide grooves (pos. *d*, Fig. 4.9) along which the pusher guides slid (pos. *e*, fig. 4.9), providing them with a trajectory of movement in which one valve lift was carried out in two revolutions of the crankshaft. It should be noted that in this engine, in addition to valve timing, the sub-piston cavity was used to pre-compress air before it was supplied to the working cylinder through an additional automatic valve mounted on the piston (pos. *g*, Fig. 4.9). Thus, this engine can also be considered the first engine on which an attempt was made to organize supercharging.

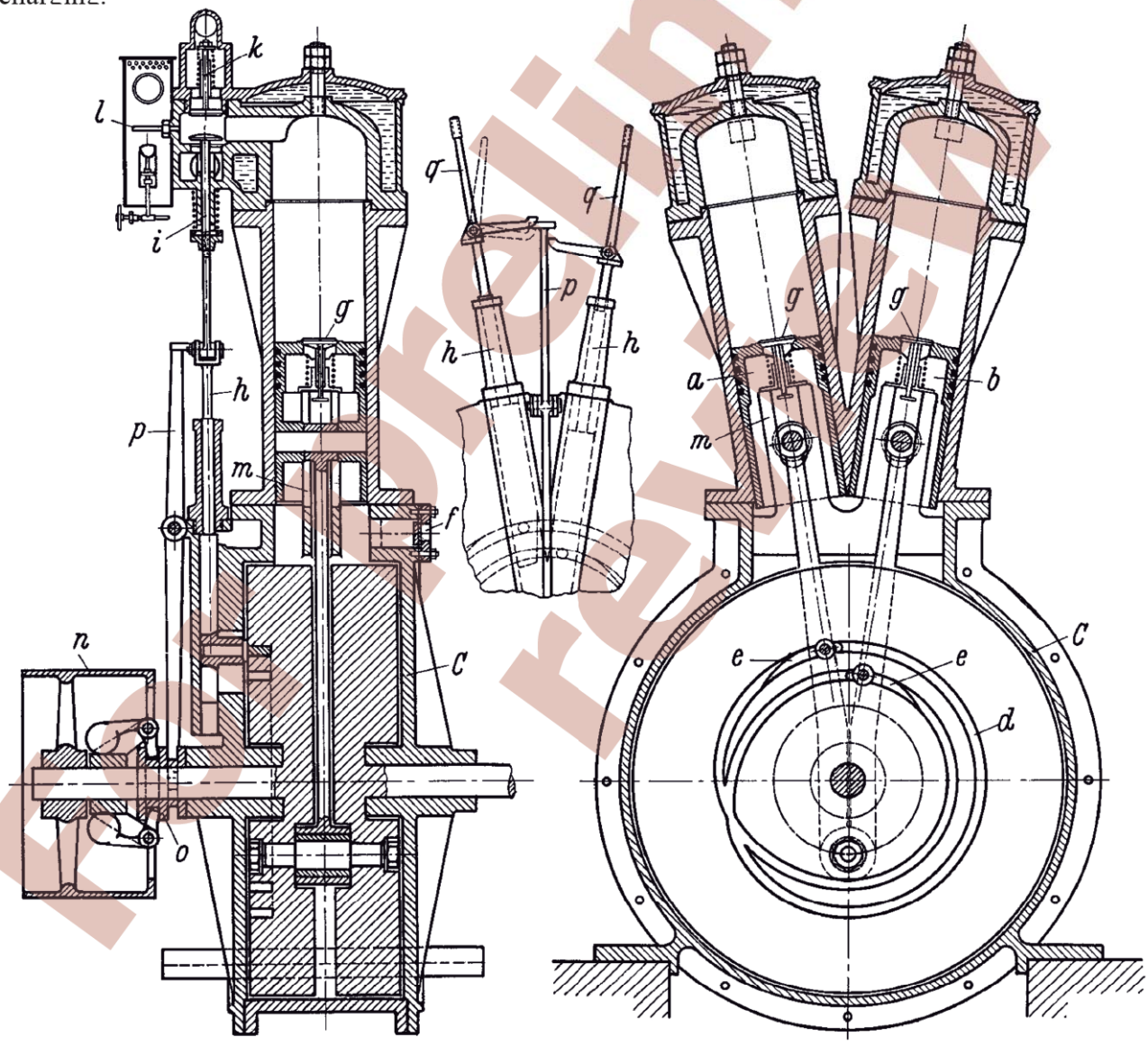


Figure 4.9 – Daimler engine with valve timing (adapted from [4])

Preliminary review

The appearance of a cam-type camshaft on engines used to drive valves did not turn out to be fundamentally new. As engine speeds increased, the speed of the automatic intake valve turned out to be too slow. On the one hand, in order to ensure rapid closing of the valve, it was necessary to increase the rigidity of the spring holding it, but this led to an increase in the resistance to its opening. As a result of this, the intake valve soon acquired a mechanical drive, as did the exhaust valve.

Before the invention of the diesel engine, a large number of different valve train designs were developed, mainly with lower valves.

A lower valve design with a side valve arrangement, in which the camshaft and valves were located in a block, in a row on the side of the cylinders, with the valve plate facing up. The drive was carried out through a pusher directly from the camshaft located underneath.

A variation of the lower valve design were engines with a T-shaped head, which were somewhat widespread. In them, the intake valves were on one side of the cylinder block, and the exhaust valves on the other. There were also two camshafts.

The main advantages of the lower valve circuit are low noise, ease of manufacture and maintenance, and the absence of danger of touching the valves and piston. All parts of this type of gas distribution mechanism are located inside the block, which makes it possible to obtain a very compact engine. The camshaft is located in a common crankcase with the crankshaft, which simplifies the lubrication system and increases reliability; there are no intermediate transmission links between the camshaft cams and valves. The head of an overhead valve engine is a simple plate with passages for coolant.

The disadvantages of a lower valve engine are that the combustion chambers have a complex shape. The need to ensure a minimum distance between the axes of the cylinder and the camshaft and the required gap between the valve plate and the walls of the combustion chamber forced the designers to give the combustion chamber a highly elongated shape. As a result, the combustion chamber had a fairly large volume, which in turn did not allow increasing the compression ratio. In practice, the compression ratio of lower valve engines did not exceed 7.5:1, which limited the possibilities for increasing their efficiency.

Obviously, it was the impossibility of obtaining high compression ratios necessary to obtain stable self-ignition of fuel in the working cylinder that forced the German engineer Rudolf Diesel to switch to overhead valve gas distribution when developing a rational engine (Fig. 4.10).

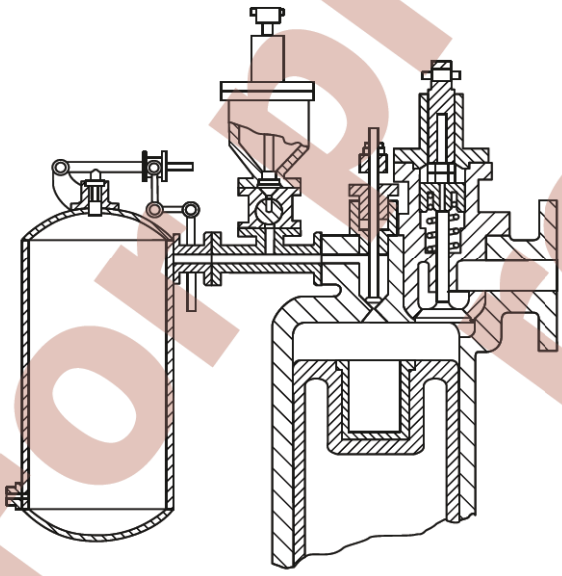


Figure 4.10 – Diesel engine with overhead valve timing (US Patent No. 608845) (adapted from [4])

Valves located in the cylinder head and opening into the cavity of the working cylinder made it possible to significantly reduce the volume of the combustion chamber and increase the compression ratio of piston engines. Thanks to this, today this scheme is the most common on all types of engines produced all over the world. The overhead valve arrangement on gasoline engines became widespread after it was used in 1904 by David Dunbar Buick* on a two-cylinder engine with a displacement 2,6 *l* and power of 22 h.p. specially designed for the Buick B car.

For preliminary review!

Along with the improvement of valve mechanisms, at the inception of engine construction, many developers tried to improve spool valve timing. Thus, at the beginning of the twentieth century, along with the already mentioned flat spool mechanisms, the so-called sleeve gas distribution was considered promising. For the first time, such a scheme with sliding sleeves was developed by the American engineer Charles Yale Knight* in 1903, and is often called the «Knight system» after his name. Knight's engine used two concentric reciprocating moving sleeves. Each sleeve was equipped with large ports on one edge. As the liner was moved up and down, these cutouts periodically aligned with the inlet or outlet port in the side wall of the cylinder. The liners were driven by an intermediate eccentric shaft, which replaced the cam shaft, which rotated at half the speed of the crankshaft. This mechanism worked very well in engines with relatively low power, but when trying to obtain high litre power, engines with a double sleeve often failed due to oil starvation of the developed friction surfaces, and therefore they were subsequently abandoned. In England, work on engines with liner gas distribution was carried out by Gary Ricardo*, who spent thirty years conducting research on sleeve gas distribution. In the course of these and similar works, versions of engines were developed in which the liners rotated around the cylinder rather than sliding along it, engines with cylinders whose ports were located in the head of the block, and not in the side walls, etc. Thanks to the work carried out, many designs of sufficiently powerful gasoline piston engines for aviation needs were created, and only the advent of gas turbine engines led to the curtailment of these works.

The first two-stroke reversible marine engine was built by Sulzer in 1905. This was the engine in which a direct-flow valve timing scheme was first used, in which intake was carried out through valves located in the cylinder cover, and exhaust through exhaust ports cut into the lower part of the working cylinder sleeve. Having a cylinder diameter 175 mm and piston stroke 250 mm, in a four-cylinder version the engine developed a power of 74 kW at 375 min^{-1} (Fig. 4.11).

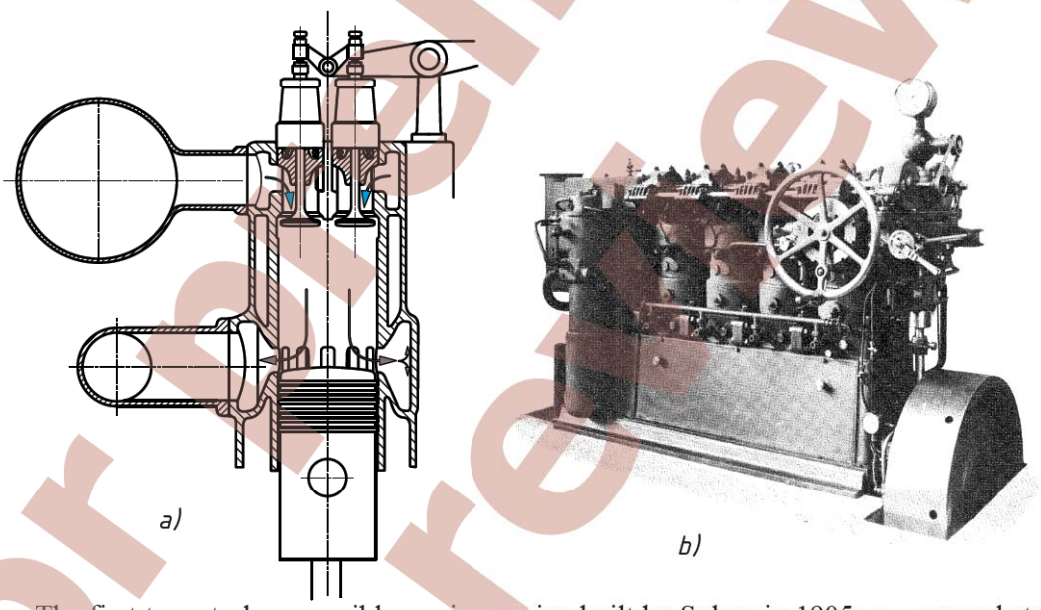


Figure 4.11 – The first two-stroke reversible marine engine built by Sulzer in 1905: *a* – general structure and gas exchange diagram; *b* – general view (adapted from [5])

Another type of spool gas distribution is a scheme in which the functions of the spool are performed by a working piston, which, when moving along the sleeve, sequentially opens or closes the ports for releasing exhaust gases and admitting a fresh charge. Sulzer developed such engines, which completely abandoned the use of valves, by 1909. The first ocean-going vessel to use such engines was the freighter «Monte Penedo», which was equipped with two four-cylinder Sulzer 4S47 crosshead bore 470 mm and stroke engines 680 mm (Fig. 4.12). The engine featured transverse loop blowing through two rows of ports cut in the lower part of the cylinder liner. At a rotation speed of 160 min^{-1}, each engine developed a power of 625 kW.

Focusing on the production of powerful marine diesel engines, the company considered the two-

For preliminary review

stroke cycle as the most rational way to increase specific power and reduce dimensions, and the absence of valves also as a way to increase reliability. The company adhered to this concept for the next 70 years.

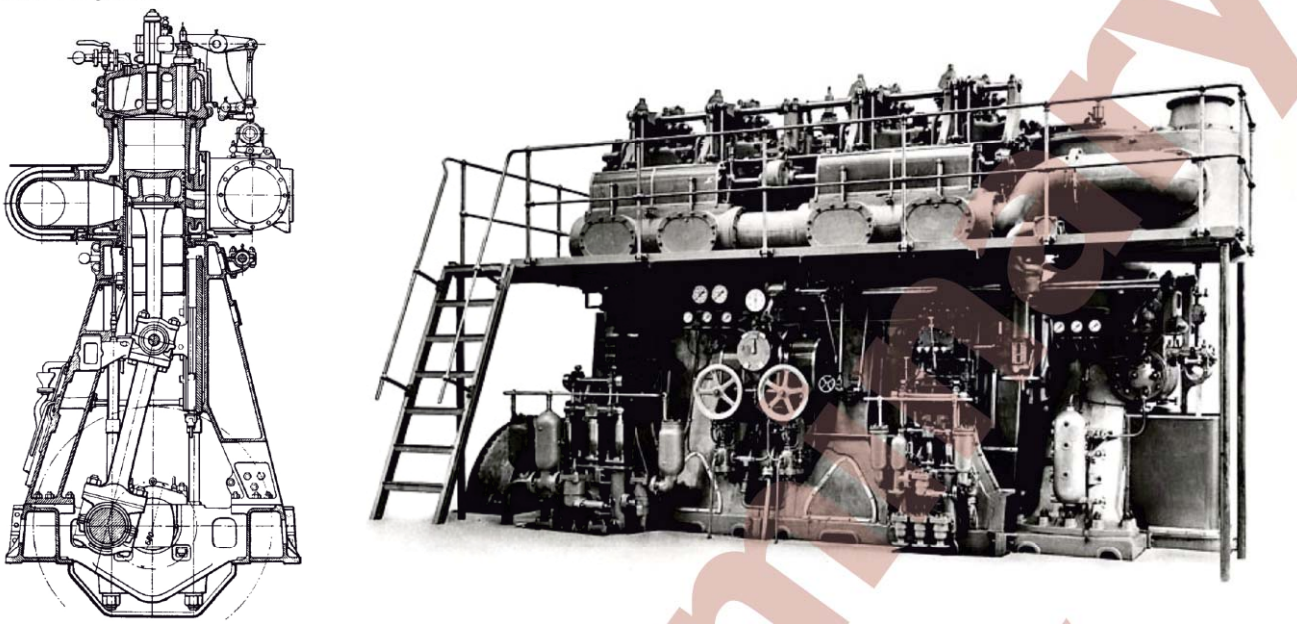


Figure 4.12 – The first two-stroke crosshead engine Sulzer 4S47 with spool valve timing installed on the ocean vessel «Monte Penedo» (1912) (adapted from [6])

A significant contribution to the development of spool valve gas distribution circuits was made by the outstanding German engineer and inventor Hugo Junkers* (Junkers Hugo). He implemented a direct-flow purge scheme in diesel engines with oppositely moving pistons created for aviation needs. In Junkers engines, one piston controlled the opening of the intake ports, and the other controlled the opening of the exhaust ports. Subsequently, such a gas distribution scheme was called the Junkers scheme. This scheme is still used today on engines for land and sea transport. In particular, the Doxford company used direct-flow blowing in its engines with opposed pistons, made according to the Junkers scheme (Fig. 2.27). The first single-cylinder engine with 368 kW was tested in 1914. In 1921, a four-cylinder engine with a bore 500 mm and a total stroke of two pistons 2320 mm, developing a power of 2200 kW, was installed on the «Yngaren» engine ship, becoming the first engine in a whole series of successful Doxford engines.

One of the ways to improve engine efficiency is to control the process of filling the working cylinder and exhaust gases. The most effective way to influence engine operating process parameters is by controlling gas exchange processes. Using a variable valve timing system that adjusts the opening, closing moments and valve stroke in accordance with the rotation speed and engine load. By changing the gas exchange parameters, it is possible to change the compression and expansion parameters, thereby achieving more efficient use of engine power, reducing fuel consumption, and reducing the toxicity of exhaust gases.

The idea of controlled gas exchange, compression and expansion was first proposed in 1882 by British engineer James Atkinson*. The essence of the idea was as follows: the efficiency of a piston engine can be increased by reducing the compression stroke and increasing the expansion stroke of the working fluid. As a result, compression costs will decrease and the useful work done by the gases in the working cylinder will increase. In gasoline engines, this also made it possible to use fuels with relatively low knock resistance.

In his patents, Aktinson proposes to implement this principle through rather complex mechanisms for driving the working piston. In Fig. 4.13 shows the design, the scheme of which is borrowed from American patent No. 367496. Several versions of the Actinson engine were built, but the increase in efficiency was largely offset by the increase in mechanical losses and the complexity of the drive mechanism, and a decrease in cylinder filling led to a loss of power compared to similar engines of other designs. For this reason, Actinson engines were not widely used.

Preliminary review

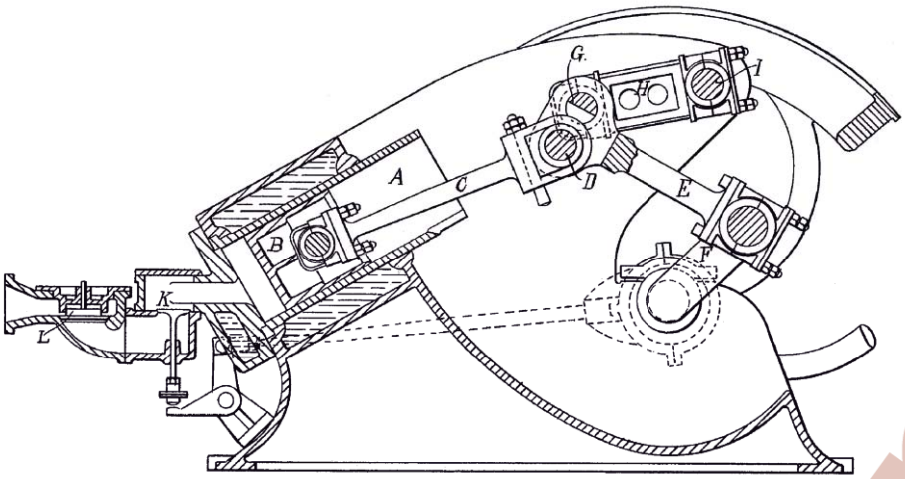


Figure 4.13 – Actinson engine scheme (borrowed from American patent No. 367496) (adapted from [4])

In 1947, American engineer Ralph Miller returned to the idea of using reduced compression and continued expansion. Unlike the Actinson engine, he proposes to implement this principle not through the kinematics of the piston drive, but by selecting the valve timing of an engine with a conventional crank mechanism.

Λ similar organization of the operating process is applicable to both engines with self-ignition from compression and engines with forced ignition. The theoretical cycle of the spark-ignition Miller engine is fully consistent with the Actinson cycle.

It became possible to control the gas exchange process most effectively with the introduction of microprocessor control systems in engine building. To actuate the valves in such systems, an electric (solenoids), hydraulic or pneumatic actuator is used.

The first low-speed electronically controlled engines were introduced into service by Sulzer in 2001 and by MAN in 2003. The goal of introducing electronic control was to optimize the operating process of engines, reduce harmful emissions from exhaust gases and reduce specific fuel consumption. The use of electronic systems has made it possible to increase flexibility in controlling gas exchange parameters, as well as the fuel injection advance angle and the law of its supply. This, in turn, made it possible to optimize engine operation over the entire range of its operating modes.

4.5.2 Types of gas distribution mechanisms used

The gas distribution mechanism serves to control the processes of gas exchange between the engine working space and the environment in accordance with the accepted phases of gas exchange (Fig. 2.9, 2.19-2.21). The following types of gas distribution mechanisms have become widespread in marine internal combustion engines:

- spool type;
- valve type;
- combined valve-spool type.

4.5.3 Spool valve timing

Spool valve gas distribution is characteristic primarily of two-stroke low-speed diesel engines. At present, engines with spool timing are not built, but in operation you can still find diesel engines with slotted purge schemes, in which the role of the spool is played by a piston that alternately opens and closes the exhaust and purge ports (Fig. 2.21).

The main advantages of such schemes are the simplicity of the design, which is associated with the complete absence of gas distribution valves and their drive. In addition, the design of the cover allows the nozzle to be positioned along the axis of the working cylinder, providing the most favourable conditions for mixture formation in the combustion chamber.

The disadvantages of spool mechanisms include the following:

- this scheme does not provide high-quality cleaning of the cylinder from combustion products remaining from the previous cycle;
- the need to ensure sequential opening of the exhaust and purge ports forces them to be located at different levels or in two rows, which leads to a significant loss of power stroke on the gas ex-

For preliminary review

change organs;

- the magnitude of the working stroke of the piston is limited, since, being in the TDC position, the piston, with its throne, must block at least the exhaust ports. As a result, the height of the piston and bushing increases, which leads to an increase in the height dimensions of the engine;

- an increase in the above-piston space leads to deterioration of cleaning conditions, especially in the upper part of the working cylinder;

- the release of hot gases through the ports led to significant thermal stresses in the liner material, especially in the area of the jumpers between the ports.

In Fig. 4.14 shows the three most common spool gas distribution schemes.

The diagram shown in Fig. 4.14 *a*, has a number of ports located along the entire generatrix of the sleeve, while the exhaust ports have a greater height and, therefore, open earlier, as a result of which the pressure in the working cylinder is relieved. When the piston is further lowered, purge ports open, having a channel inclined to the cylinder axis and located eccentrically in plan. This arrangement ensures the movement of incoming air from the purge ports to the exhaust ports, and therefore this scheme is called *cross-slot*. The main disadvantage of this scheme is that the location of the purge ports opposite the exhaust ports leads to the danger of «circuiting» the air flow, when a significant part of the air moves along the shortest path – from one port to another, without cleaning the entire volume of the cylinder.

The diagram shown in Fig. 4.14 *b* allows you to eliminate the opposite arrangement of purge and exhaust ports. This scheme and similar ones are usually called *loopback*. This scheme eliminates the possibility of circuiting of gas-air flows in the cylinder, which mix with each other less intensively. At the same time, the volume of stagnant zones in the cylinder increases, which leads to a deterioration in its cleaning. With this arrangement, the inlet and outlet receivers are located on one side of the engine, which reduces its width dimensions (Fig. 4.14).

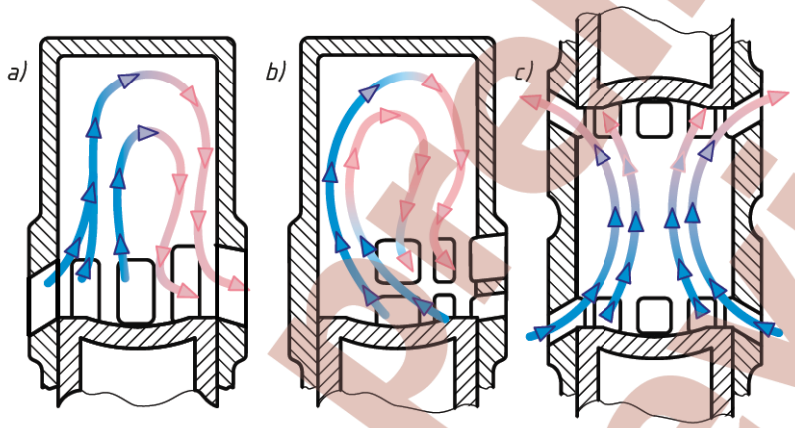


Figure 4.14 – Slot blowing schemes: *a* – transverse slot; *b* – loop; *c* – direct-flow slot

The diagram shown in Fig. 4.14 *c*, devoid of most of the disadvantages inherent in the previous two. This scheme, implemented in two-stroke engines with counter-moving pistons, is called *direct-flow slotted.* The purge and exhaust ports in such engines are located at different ends of the working cylinder in close proximity to the BDC of the pistons. When the ports are opened with pistons, the air moves along the axis of the cylinder, ensuring its good cleaning. To prevent exhaust gases from being thrown into the intake receiver, the exhaust ports open earlier than the purge ports. To achieve this, the piston phases are slightly offset relative to each other. The main disadvantage of this scheme is the significant complication of the design, and first of all, of the crank mechanism.

For a long time, the advantages of spool valve timing were decisive. In the process of improving gas exchange schemes, different relative positions of the ports and their orientation relative to the cylinder axis, different ratios of the heights of the exhaust and purge ports were used; spools, dampers and valves were installed on the exhaust and purge ports. However, numerous attempts to get rid of the shortcomings of this scheme have not yielded positive results. For this reason, all leading manufacturers have abandoned the production of this type of engine. The only exceptions are engines with a direct-flow-slot design, according to which marine two-stroke MSE and HSE are still produced.

For preliminary review

4.5.4 Valve timing

Valve mechanisms is most widespread on marine four-stroke engines due to its comparative simplicity of design and reliable operation. In engines of this type, the opening and closing of the intake and exhaust ports is carried out using valves located in the cylinder cover.

On the Fig. 4.12 shows the general structure of the valve gas distribution mechanism, which assembly of valve units and a transmission mechanism.

The valve assembly includes: valve 1 (inlet or exhaust), valve seat 2, guide sleeve 3, spring 4 and its lower and upper plates 5.

The transmission mechanism in most cases consists of: levers 6, rods 7, pushers 8 and a camshaft with a set of cam washers 9.

The valves are lifted by the action of the camshaft cams, and seated and held closed by the valve springs.

In a number of four-stroke engine designs, a hydraulic transmission is used to drive the valves, in which the drive cam acts through a pusher on the hydraulic drive pump plunger called *an actuator.* The actuator creates oil pressure in the valve drive system, under the influence of which the actuator pistons move down, opening the valves (Fig. 4.16). Otherwise, this mechanism is similar to that

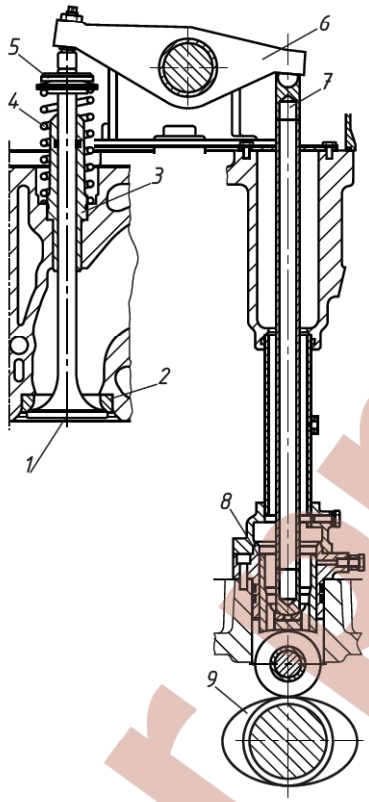


Figure 4.15 – Valve-type gas distribution mechanism of diesel engine TBD 645, Deutz MWM: 1 – valve; 2 – valve seat; 3 – guide sleeve; 4 – spring; 5 – upper valve plate; 6 – double-arm lever; 7 – pusher rod; 8 – pusher; 9 – camshaft (adapted from [7])

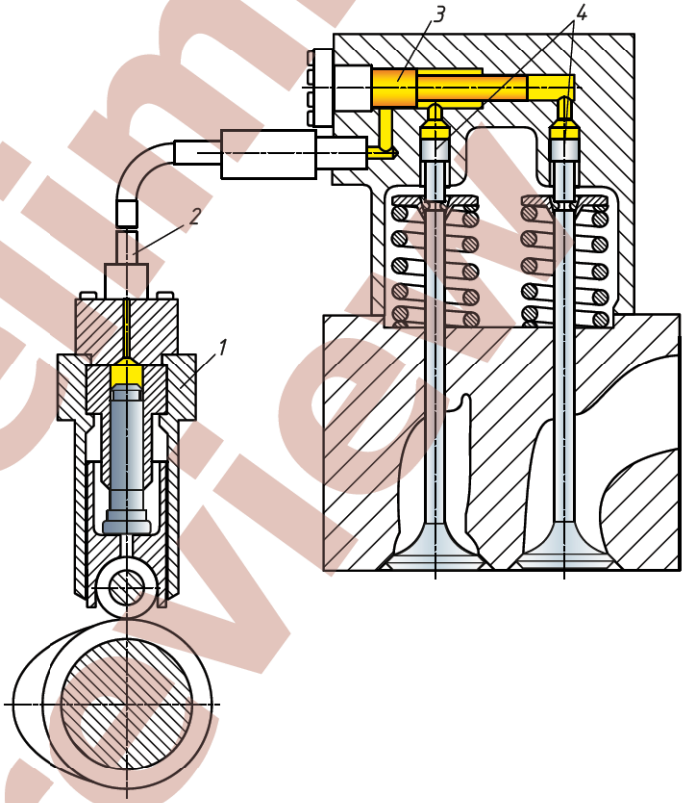


Figure 4.16 – Diagram of the gas distribution mechanism with hydraulic valve drive of the Sulzer ZA50S engine: 1 – actuator, 2 – high-pressure oil line; 3 – oil filter; 4 – valve drive pistons (adapted from [6])

The layout of the gas distribution mechanism shown in Fig. 4.12 and 4.13 is typical for most modern marine internal combustion engines. The placement of valves with this arrangement is called *top*, and the valves are called *suspended.* The upper location of the valves allows for a compact combustion chamber, which helps reduce heat losses and improve mixture formation, especially with a central location of the fuel injector.

The quality of cleaning and filling the cylinder, other things being equal, depends mainly on the size of the flow area of the gas distribution elements. The size of the flow area, in turn, depends on

For preliminary review

the diameter of the valves and their number per cylinder. With a two-valve design, which involves installing one intake and one exhaust valve, the increase in valve diameter is usually limited by the dimensions of the cylinder. To increase the flow area, most modern engines are made using a four-valve design, which involves installing two intake and two exhaust valves on one cylinder. This approach makes it possible to more rationally use the area of the fire bottom for gas exchange organs; it is enough to simply arrange the cylinder cover with a central location of the fuel injector and improve heat removal from the valves.

4.5.5 Combined valve-spool valve timing

This type of gas distribution is predominantly used on two-stroke engines with direct-flow valve scavenging. In almost all modern LSEs, as well as some types of MSE and HSE, the working cylinder is purged through purge ports located in the lower part of the cylinder liner. The opening of the ports is controlled by a working piston, which simultaneously performs the functions of a spool. Exhaust gases are released through exhaust valves located in the cylinder cover and driven from the engine crankshaft through a drive mechanism. The general diagram of purge of the working cylinder of a two-stroke engine with direct-flow purge and valve-spool gas distribution mechanism is shown in Fig. 4.17.

For LSE, gas distribution mechanisms with one central valve per cylinder are most widely used. This design makes it possible to significantly simplify the design of the valve assembly and drive mechanism, as well as to maximize the flow area of the valve slot. Its main disadvantage is the impossibility of central location of the fuel injector. In order to evenly distribute the fuel aerosol throughout the volume of the combustion chamber, two or three peripheral injectors are installed on engines made according to this design, which leads to a more complex fuel system. On two-stroke high-speed trunk engines, a common design is in which the fuel injector is located in the center of the fire bottom of the cylinder cover, and the valves are located on the periphery of the cylinder axis. For example, in engines of the EMD series from General Motors, one central injector and four exhaust valves with a synchronous drive are installed in the cylinder cover (Fig. 4.18).

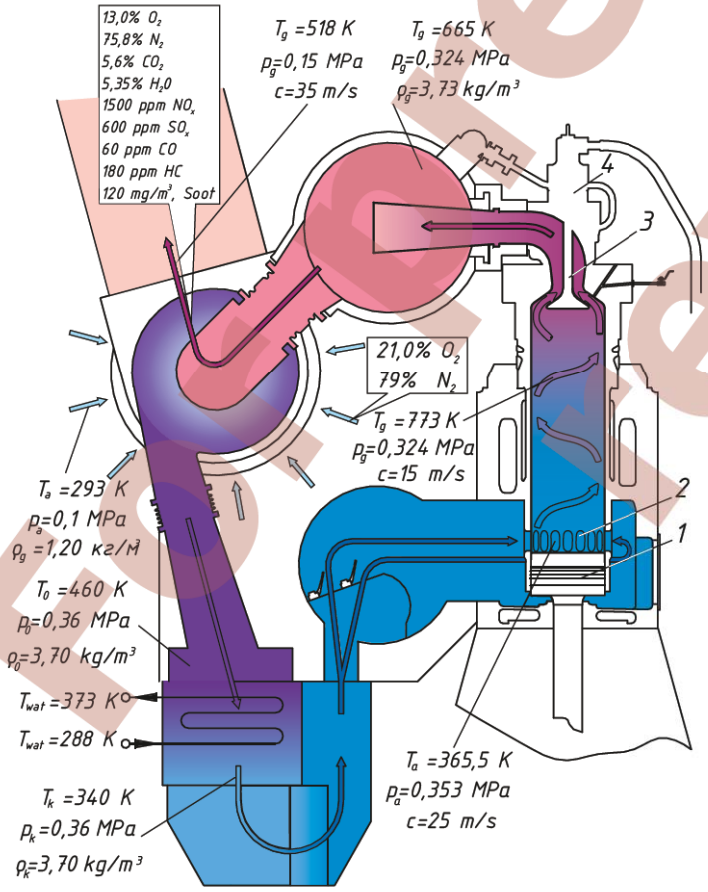


Figure 4.17 – Scheme of direct-valve purge of a two-stroke LSE with a valve-spool gas distribution mechanism and parameters of the working fluid at different stages of the cycle: 1 – piston performing the functions of a spool; 2 – purge ports; 3 – exhaust valve; 4 – valve drive mechanism (adapted from [8])

Figure 4.18 – Cylinder cover of General Motors EMD series engines with one central injector and four exhaust valves (adapted from [9])

Preliminary Preview

On four-stroke marine engines, the valve-spool arrangement of the gas distribution mechanism is used extremely rarely. An example of its use is the design used by the Japanese company Mitsui in the ADD30V series engines. The general layout of the gas distribution mechanism and the principle of its operation are shown in Fig. 4.19.

In the cylinder cover of this engine, the intake and exhaust cavities are located one above the other in two tiers. Between the cavities there is a cylindrical bore in which a spool is installed in the shape of a hollow cylinder made of heat-resistant steel. Through the spokes in the internal cavity of the spool, its cylindrical part is connected to the drive mechanism (Fig. 4.19 *a*).

On the working cylinder side, the gas exchange cavities are closed by a central valve, called the *main valve.*

During the compression and working strokes, the main valve is closed, separating the working space of the cylinder and the gas exchange cavity (Fig. 4.19 *b*). The exhaust stroke corresponds to the lower position of the spool, at which its cylindrical part blocks the inlet cavity of the gas tract. When the main valve opens shortly before the piston reaches BDC, gases pass through the internal space of the spool into the outlet cavity located in the second tier (Fig. 4.19 *c*) and then enter the exhaust manifold. As the piston approaches TDC, under the action of the drive mechanism, the spool rises to the top, blocking the outlet cavity with its end surface and opening the inlet cavity, located in the first tier (Fig. 4.19 *d*). As a result, fresh charge begins to enter the cylinder through the open main valve. After the piston passes BDC, the main valve begins to close, as a result of which the intake process gradually ends. After the valve is seated, the spool moves to the bottom, taking its original position (Fig. 4.19 *e*).

The opening of the main valve and the upward movement of the spool occurs under the action of a hydraulic drive, and the reverse movement occurs under the action of a gas spring. Air enters the cavity of the gas spring from the engine air-starting system.

This scheme has a number of advantages over the classic valve timing, which include:

- the presence of one valve makes it possible to ensure the maximum flow area of the valve slit;
- the spool is unloaded from the high pressures acting in the working cylinder;
- alternately washing the main valve with hot and then cold gases allows you to reduce its thermal intensity.

Disadvantages include difficulties with the layout of the combustion chamber and the need, therefore, to install two injectors per cylinder.

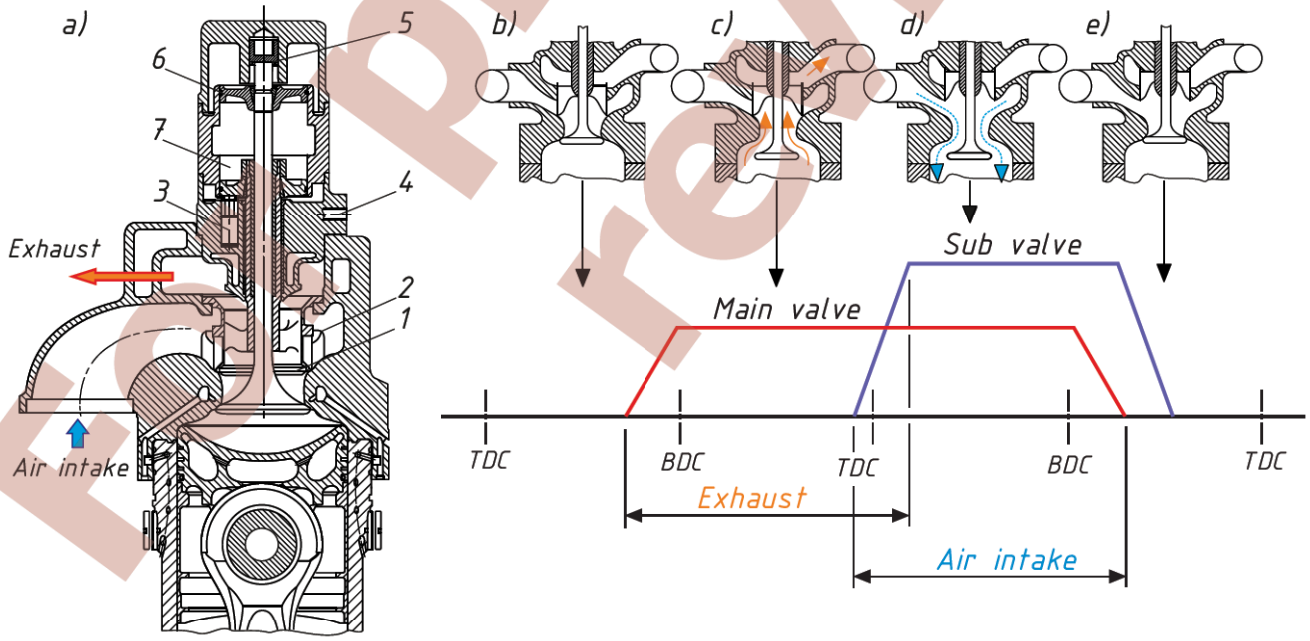


Figure 4.19 – Spool-valve timing mechanism of a four-stroke ADD30V engine from Mitsui: 1 – spool; 2 – inlet cavity; 3 – pistons of the hydraulic spool drive; 4 – oil inlet to the hydraulic spool drive; 5 – working cavity of the main valve hydraulic drive piston; 6 – main valve; 7 – air cavity performing the functions of a gas spring (adapted from [6, 10])

For preliminary review

4.5.6 Assemblies and elements of valve and valve-spool gas distribution mechanisms

Valve assemblies of low-speed engines with direct-flow valve purge are installed directly on the cylinder cover (Fig. 4.20).

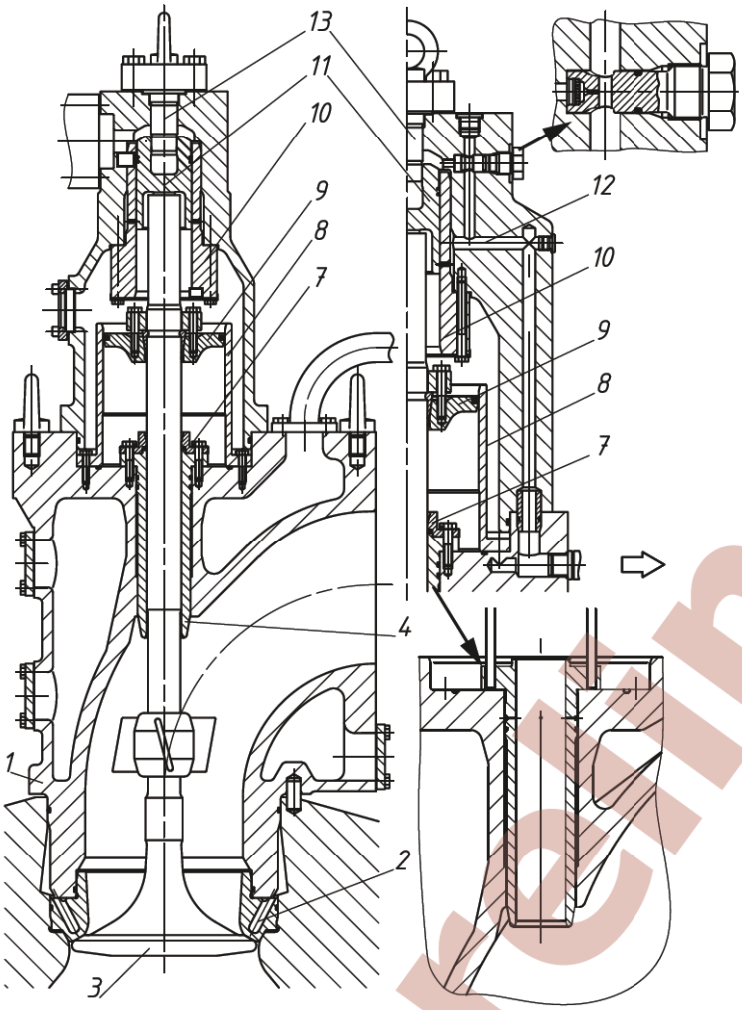


Figure 4.20 – Valve assemblies of low-speed diesel engine RTA 72 U from Wärtsilä with a hydropneumatic valve drive. 1 – housing; 2 – valve seat; 3 – valve; 4 – valve guide bushing; 5 – springs; 6 – spring plate; 7 – valve stem seal; 8 – working cylinder of the pneumatic spring; 9 – pneumatic piston; 10 – hydraulic cylinder of the valve drive; 11 – hydraulic piston of the valve drive; 12 – drainage channel for collecting leaks; 13 – valve hydraulic brake stop (adapted from [11])

They consist of a housing, a valve seat, the valve itself, its guide sleeve, a gas spring holding the valve in the closed position and plates for fixing them (Fig. 4.20). Modern low-speed engines use valve units, which, in addition to the elements listed above, include a valve drive mechanism. Mostly, the valves of modern LSEs open under the action of a hydraulic drive, and are closed and held closed by a gas spring. The air in the drive gas chamber comes from the ship's general air supply system. The hydraulic fluid for the valve drive comes from an actuator pump driven from the engine camshaft or having a hydraulic drive.

For four-stroke MSE and HSE valve assemblies are usually made directly in the cylinder cover housing (Fig. 4.21 *a*, *d*). In some engine models, they are manufactured in separate housings (Fig. 4.21 *c*).

The intake and exhaust valve plates are usually made the same, however, for some HSEs, for better filling of the cylinder, the diameter of the intake valve is increased by 15...20% compared to the exhaust. The main sealing surface of the valve disc is an annular chamfer, which is made at an angle of 45° for the exhaust valve and 30...45° for the inlet valve. This chamfer presses tightly against the valve seat. During operation, the valves are subject to mechanical loads from gas pressure forces and inertia forces, as well as significant heating. The high temperature and velocity of gases containing such aggressive components as sulfur and vanadium compounds cause corrosion and erosion of valve seating surfaces and their seats. To increase service life and more uniform temperature distribution, the valves are given a rotational movement using a mechanism called a *rotocap* (Fig. 4.21 *f*) or an impeller (Fig. 4.21 *h*) called an *impeller*. In some engines, to increase wear resistance, a layer of a hard wear-resistant alloy called *stellite* is deposited on the sealing chamfer of the exhaust valves.

For preliminary review

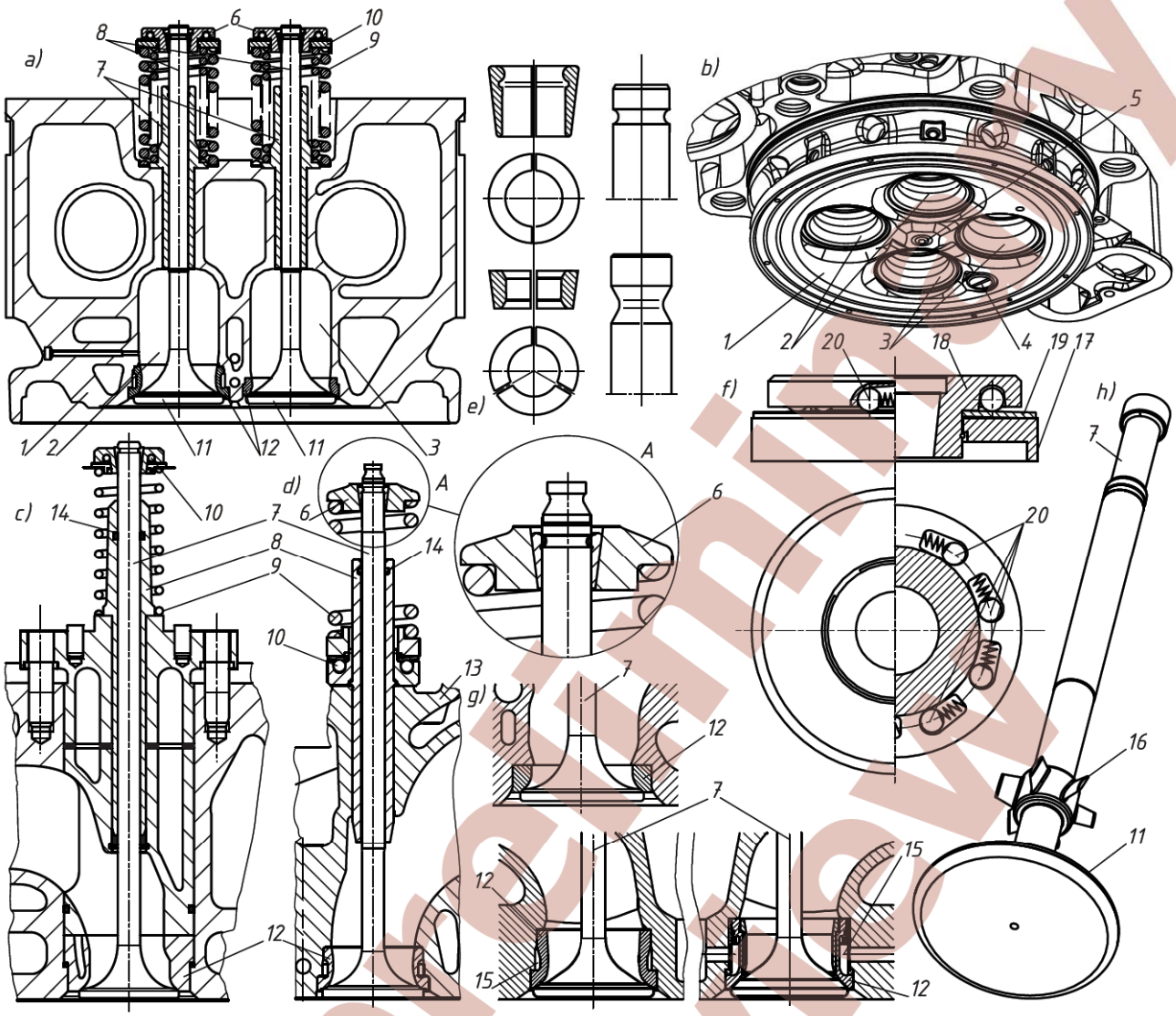


Figure 4.21 – Valve gas distribution mechanisms and their elements: *a* – cylinder cover with suspended valves of the 46 series diesel engine from Wärtsilä; *b* – fire bottom of the MaK 43C diesel engine with valve sockets; *c* – plug-in valve housing for diesel engine TBD 645, Deutz MWM; *d* – valve assembly of the MaK 25C diesel engine; *e* – conical cotters for fastening the upper valve plate; *f* – device for turning the «rotocap» valve; *g* – plug-in valve seats; *h* – exhaust valve of MAN K98MC-C diesel engine:: 1 – firing bottom; 2 – exhaust valve sockets; 3 – intake valve sockets; 4 – start valve; 5 – fuel injector spray tip; 6 – upper valve plate; 7 – valve stem; 8 – valve guide bushing; 9 – spring; 10 – rotocap; 11 – valve plate; 12 – valve seat; 13 – cover housing; 14 – sealing ring; 15 – water cooling cavity of the exhaust valve seat; 16 – impeller; 17 – rotokap stator; 18 – rotokap rotor; 19 – conical spring washer; 20 – spring-loaded balls (adapted from [7, 12, 13, 14, 15])

In the upper part of the valve stem there is a groove for installing conical wedge-shaped fasteners called *crackers* (Fig. 4.21 *e*), which fit into a cylindrical or conical groove on the housing of the rod. With their help, a valve spring thrust washer called *a plate is attached to the shank of the rod* (Fig. 4.21 *d*).

The valve spring ensures that the valve closes and presses tightly against the seat when closed. One end of the spring rests on the valve housing or cover, and the other rests against the upper valve plate. On most modern MSEs and HSEs, rotary caps are installed under the springs, mounted in the upper plate of the valve spring (Fig. 4.21 *a*) or installed between the spring and the supporting surface of the cylinder cover or valve assembly housing (Fig. 4.21 *d*).

In the mounted state, the spring must have a stiffness sufficient to ensure kinematic connection

For preliminary review

between the elements of the valve drive mechanism. In marine internal combustion engines, the most widespread are cylindrical coil springs with a constant winding pitch.

To prevent serious accidents associated with spring breakage, as well as to ensure the necessary rigidity, two springs of different diameters, located concentrically, are usually installed on one valve. To eliminate the possibility of the coils of one spring getting between the coils of another, they are wound in opposite directions (Fig. 4.21 *a*).

On push-pull LSEs previously used peripheral springs located along the valve stem. This design reduces the likelihood of valve assembly failure if one of the springs breaks, and simplifies the technology for their maintenance and dismantling.

The springs are centered relative to the valve stem using mounting pins on the valve plates. Valve springs are made from special spring wire using the cold coiling method.

The valve guide is usually removable. To reduce friction between the rod and the guide surface in conditions of limited lubrication, bushings are made of bronze or cast iron alloys. On the outer surface of the bushing there is a special limiting collar, along which it is pressed into the valve housing or directly into the cylinder cover (Fig. 4.21 *a*, *c*, *d*, *h*).

On LSE with direct-flow valve purge schemes, the valve guides are fixed using a flange made in its upper part. The flange is attached to the valve assembly housing using a bolted connection (Fig. 4.20).

To limit the entry of lubricating oil into the gap between the valve and the bushing, as well as to reduce leakage of charge air, an O-ring made of rubber or special polymer materials is installed in the upper part of the bushing.

The valve seat is the supporting surface on which the valve disc sits with its sealing chamfer. The chamfers of the seat and valve are carefully ground to each other to ensure good sealing of the joint and effective heat dissipation.

Structurally, the seat can be made by boring directly into the housing of the cylinder cover or in the form of a separate removable ring element, which can be secured in the cover by pressing (Fig. 4.21 *a*, *d*, *g*) or spring rings.

The MAN company uses a seat design on its low-speed engines, the working edge of which is made with two chambers (W – shaped seating surface of the seat, Fig. 4.22). This design allows you to increase the specific pressure of pressing the valve to the seat. The gap existing between the first edge and the valve acts as a labyrinth seal, which, by throttling the gases, allows to reduce the pressure in the first chamber, thereby reducing the load on the working edges and increasing the reliability of the entire gas joint.

To reduce wear on the working edge of the valve, a hard-alloy, heat-resistant coating of chromium-nickel alloy (Inconel) is applied to its seating surface along its entire diameter by surfacing.

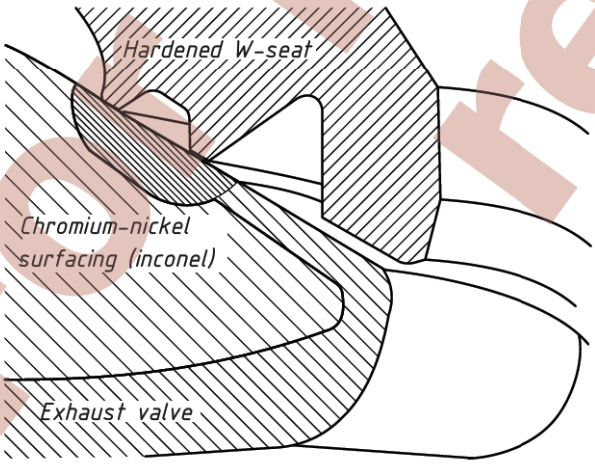


Figure 4.22 – W-shaped valve seat seating surface (adapted from [16])

In most modern marine diesel engines, in order to more intensively remove heat from the valve plate, the exhaust valve seat is made cooled by supplying water from the cooling system to it through special channels in the fire bottom of the cylinder cover. The seats are made of special cast iron or heat-resistant steel.

For preliminary review

The valve rotating mechanism is installed on most modern marin e diesel engines. For rotation, impellers are used (Fig. 4.21 *h*), which are affected by the flow of exhaust gases, or a special mechanism called a rotocap.

The rotocap (Fig. 4.21 *f*) consists of a stationary stator and a rotating rotor. A conical spring washer is installed between the stator and the rotor, on which the rotor rests through several spring-loaded steel balls. The balls are located in profiled grooves, the supporting surface of which is inclined to the plane of the rotor. When the valves are closed, the force of the valve spring is not enough to deflect the spring washer, so there is a gap between the stator and the rotor. The balls, under the action of the springs, occupy an extreme position in the groove, resting against the spring washer. When the valve is pressed, the force of the valve spring overcomes the resistance of the conical spring washer, bending it, which in turn presses on the balls, which roll to the bottom of the profiled groove, and the rotor rotates at a certain angle relat ive to the stator. When the valve is closed, the spring washer returns to its original position, and the balls, under the action of peripheral springs, occupy their initial positions.

The valves are driven by the camshaft, which rotates from the engine crankshaft through the *drive mechanism.* The transmission of forces from the camshaft to the valves is carried out through a *transmission mechanism.*

Drive mechanism. Considering that four-stroke engines have one power stroke per two revolutions of the crankshaft, the camshaft speed should be half that of the crankshaft. This is achieved by selecting the appropriate transmission ratio of the drive mechanism. For two-stroke diesel engines, the camshaft speed coincides with the crankshaft speed. The valve drive of four-stroke MSE and HSE is usually carried out through intermediate parts in the form of pushers, rods, rocker arms, traverses, which form a *transmission mechanism*. In some HSE valves are driven directly from the camshaft located in the cylinder head. This allows you to reduce the inertia forces in the drive and increase its rigidity. The presence of certain parts of the transmission mechanism depends on the location of the camshaft and the selected drive scheme.

For LSEs with mechanical valve control, the valves are opened via hydraulic transmission from the actuator pump to the valve drive actuator. The actuator itself is driven by a fist on the engine camshaft through the drive mechanism (Fig. 4.23 *a*). The valves are closed using the energy of compressed air or a return spring.

For LSEs with electronic control the actuator drive is hydraul ic, carried out from a common control line, through solenoid valves controlled using microprocessor technology (Fig. 4.23 *b*).

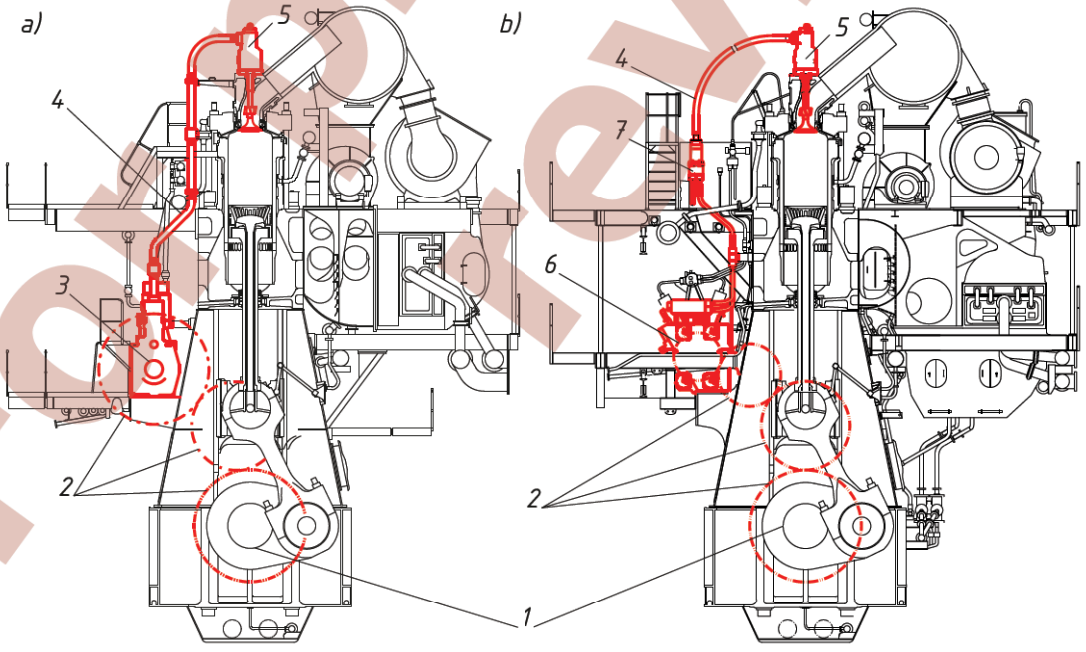


Figure 4.23 – Location of elements of the exhaust valve drive mechanism: *a* – RTA series engines; *b* – RT-flex from Wärtsilä: 1 – crankshaft; 2 – drive gears; 3 – p ump-actuator; 4 – hydraulic line; 5 – valve drive mechanism; 6 – hydraulic drive pump block; 7 – control valve (adapted from [17, 18])

For preliminary review

To drive the valves, cam washers are used, located in a certain way in accordance with the cylinder operating order adopted for a given engine. In addition to the valve drive cams, the camshafts may have cams to drive the fuel pump, mechanical indicator, lift pump, etc. In addition, a drive, a speed controller, a starting air distributor, etc. can be supplied from the camshaft. To reduce the magnitude of torsional vibrations, dampers of the type described in paragraph 3.12 can be mounted on the camshaft of some engines.

Camshafts are designed to control the movement of valves. They consist of gears or drive sprockets, a shaft housing, cam washers and bearing journals (Fig. 4.24).

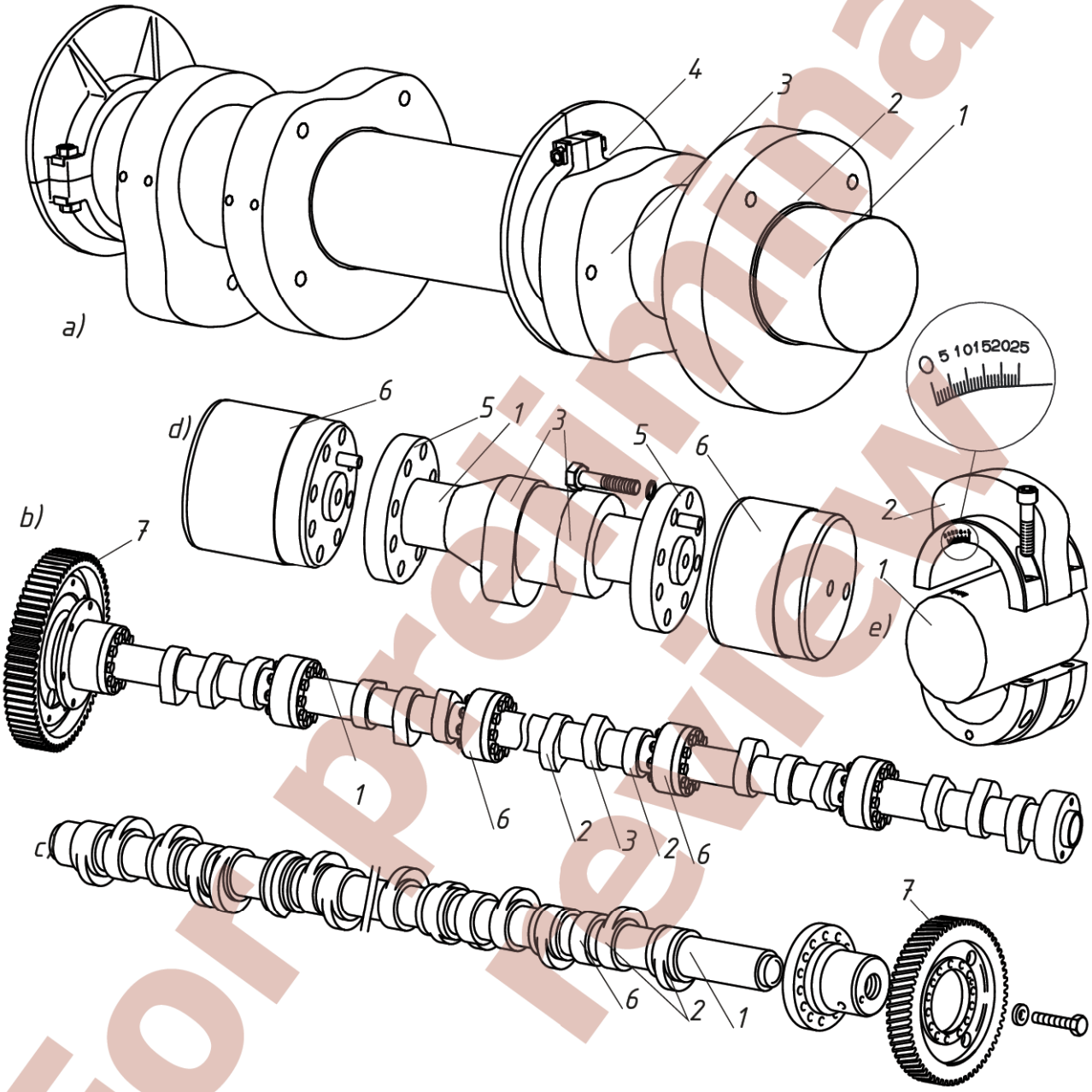


Figure 4.24 – Camshafts of marine diesel engines: *a* – two-stroke low-speed; *b* – sectional medium-speed four-stroke; *c* – one-piece high-speed four-stroke; *d* – separate section of the camshaft with bearing journals; *e* – split cam washer LSE with a scale for adjusting the position of the cam washer relative to the shaft: 1 – shaft housing; 2 – valve drive cam washers; 3 – cam washer of the fuel pump drive; 4 – cam washer of the indicator drive; 5 – connecting flanges of individual sections; 6 – supporting surface; 7 – drive gear (adapted from [19, 20, 21])

Camshafts of modern marine diesel engines, as a rule, consist of several sections (Fig. 4.21 *a*, *b*), and only the shafts of relatively small engines are sometimes made in one piece

For preliminary review

(Fig. 4.24 *c*). The individual sections are assembled using flanges secured with threaded connections. The sectional design of camshafts makes it possible to simplify the technology of their manufacture, installation and dismantling, and, if necessary, makes it possible to replace a damaged section.

The valve drive cams must ensure their rapid opening and closing with acceptable inertia forces. This is achieved by choosing the appropriate profile, which sets the law of change in the flow area of the valve slit, the magnitude of the valve stroke, the moments of its opening and closing (Fig. 4.25).

Cam profiles are usually made symmetrical. For reversible diesel engines, this makes it possible to ensure the same conditions for cleaning the cylinder in forward and reverse modes. On non-reversible diesel engines, there are designs with asymmetrical cam profiles. In four-stroke reversible diesel engines, two sets of cam washers are installed – one for forward, the other for reverse.

In low-speed and some medium-speed diesel engines, removable cam washers are installed on the camshafts (Fig. 4.25 *a*, *d*, *e*), which have a press fit with the shaft or are a split design with coupling bolts (Fig. 4.25 *e*). During a press fit to remove the cam, as well as to adjust its position, oil is pumped into the gap between the parts under high pressure, which creates an oil cushion between the surfaces that allows the cam to move relative to the shaft. To control the position of the cam, an angular scale is applied to it, and a mark is made on the shaft (Fig. 4.24 *e*). The cam for the indicator drive usually consists of two parts connected by two fitting bolts (Fig. 4.24 *a*).

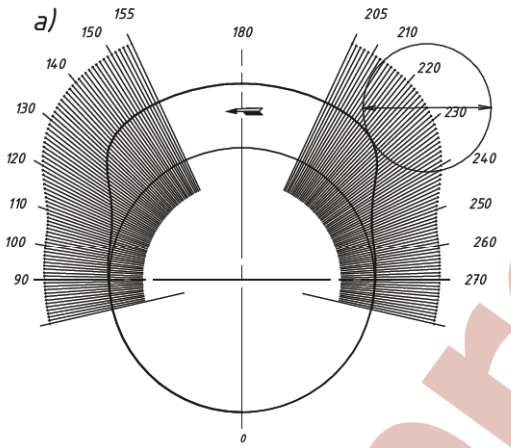


Figure 4.25 – Profiles of valve drive cams: *a* – exhaust valve of a two-stroke LSE; *b* – four-stroke MSE (adapted from [14, 19])

The camshafts of the LSE and some MSEs, together with fuel pumps, actuators and indicator drives, are assembled in the form of separate units mounted on the engine frame (Fig. 4.26 *a*). To lay the shaft in each pump block, a bed with covers is bored out, in which replaceable thin-walled support and thrust liners are installed, covered with a layer of anti-friction material. The outer liners serve to limit the axial and radial movements of the shaft, all the others – only for radial ones. A section of the shaft housing is used as a supporting surface, which is processed accordingly. As a rule, one pump unit is installed on two working cylinders.

In MSE and HSE cams and flanges are usually made as one piece with the shaft of a separate section (Fig. 4.26 *b*), connecting them directly to each other or through a cylindrical spacer insert. The developed cylindrical generatrix of connecting flanges or inserts is usually used as the supporting surface of the camshaft.

To increase rigidity, the shaft bed is usually bored into the engine block (Fig. 4.26 *b*, *c*, *d*), placing it in a special compartment for in-line and V-shaped engines or in the camber of the cylinder block for V-shaped ones (Fig. 4.26 *d*). Replaceable cylindrical bushings coated with antifriction material are installed in the bored sockets, acting as support journal bearings (Fig. 4.26 *b*). The camshafts are secured against axial movement using thrust bearings, which are usually located on the drive side. When using helical gears, this makes it possible to reduce the influence on the valve timing of thermal extensions of the camshaft and engine frame, and when using a chain drive, it avoids additional loading of sprockets and chains by axial forces.

Solid shafts are secured with caps for ease of installation. In this case, the shaft bearings are

For preliminary review

made in the form of two liners (Fig. 4.26 *c*).

Some HSEs have camshafts located in the cylinder block cover. With this arrangement, the cams act on the valves through single-arm levers (Fig. 4.26 *e*) or directly (Fig. 4.26 *f*). The upper layout of the camshaft makes it possible to reduce the number of intermediate parts of the transmission mechanism, thereby simplifying the engine design and reducing the inertia forces acting in the drive.

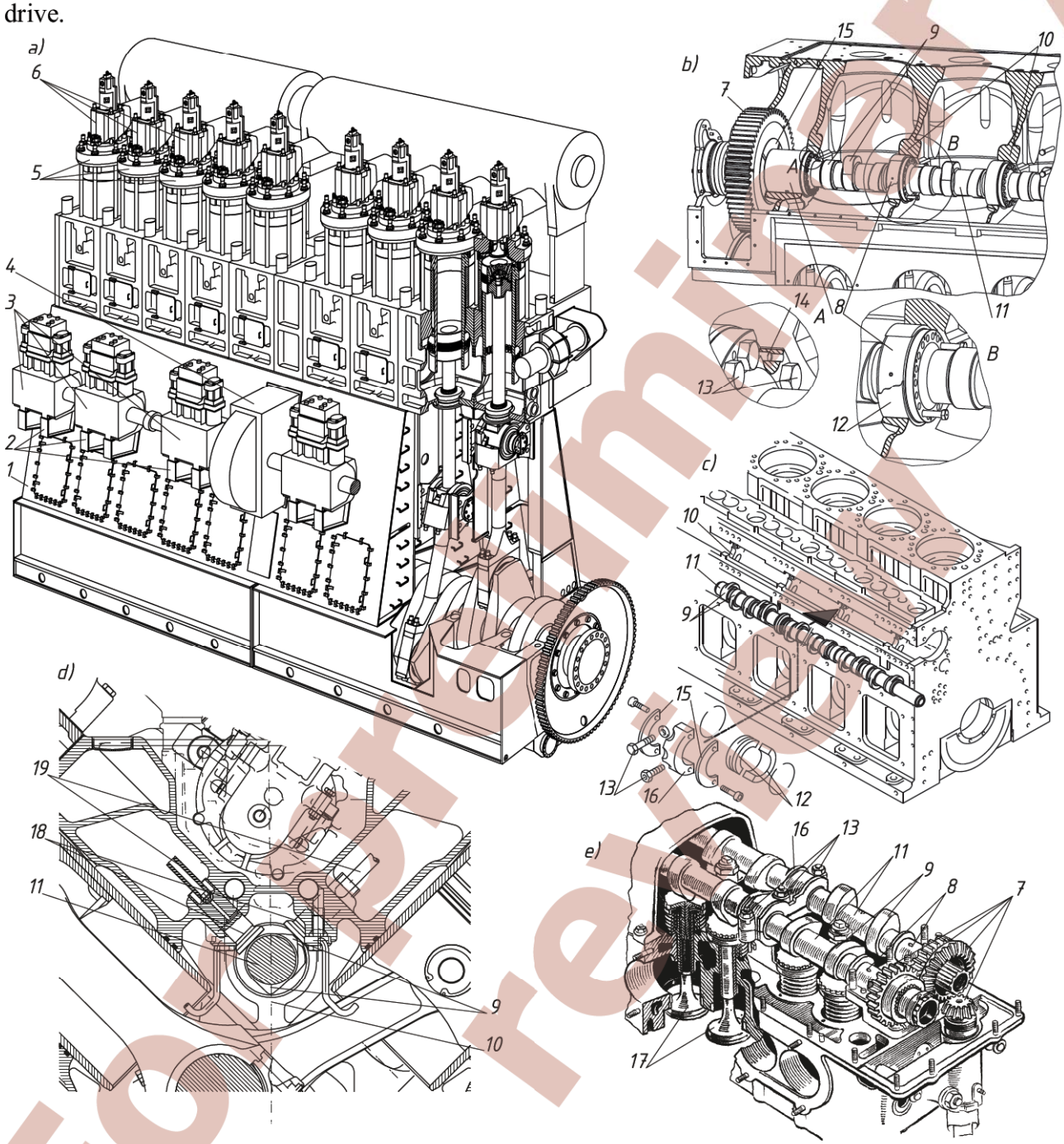


Figure 4.26 – Location of camshafts: *a* – in the form of separate units (pumping units) on the frame of a low-speed engine of the 9RTA 84C series from Wärtsilä; *b* – in the crankcase compartment of a diesel engine of the M25 series from MaK; *c* – in the crankcase compartment of a diesel engine of the TBD 645 series, made by Deutz MWM with a solid camshaft; *d* – in the camber of the cylinder block of the BF6M1015 diesel engine from Deutz MWM; *e* – in the cylinder head of the 3D6 diesel engine: 1 – crankcase; 2 – fastening shelves; 3 – pump block; 4 – drive mechanism casing; 5 – cylinder cover; 6 – valve drive mechanism; 7 – camshaft drive gear; 8 – supporting surfaces; 9 – valve drive cams; 10 – camshaft bed; 11 – shaft housing; 12 – plain bearing; 13 – fastening bolts; 14 – mounting pins; 15 – single-arm pusher lever with roller; 16 – hydraulic thermal gap compensators; 17 – valves; 18 – camshaft bed cover; 19 – thrust bearings; 20 – liners; 21 – pushers; 22 – push rods (adapted from [13, 17, 21, 22])

For preliminary review

Depending on the design, one or more camshafts may be installed on one engine. For example, on V-shaped engines, each bank of cylinders may have a separate camshaft line (Fig. 2.3). For HSEs with overhead camshafts, the intake drive shaft and exhaust valve drive shaft can be installed separately (Fig. 4.26 *e*, *f*), etc.

Camshafts are made with removable cams, usually from steel. Solid shafts are stamped from steel or cast from special cast iron.

The camshafts are driven from the crankshaft through a gear or chain transmission.

In most engines, the drive mechanism is located at the end of the crankcase. For LSEs with a large number of cylinders (usually more than seven), the camshaft drive is located in the middle part of the frame (Fig. 4.26 *a*). This makes it possible to reduce the twisting of the shafts along the length and the associated violations of the valve timing.

The choice of the type of camshaft drive largely depends on the type and speed of the diesel engine, as well as on the location of the shaft itself.

Gear drives are characterized by high operational reliability, however, as the distance between the axes of the crankshaft and camshaft increases, the dimensions of the drive increase significantly. To reduce them, install idler gears. In turn, the use of idler gears requires the installation of support shafts, which ultimately leads to a complication of the drive mechanism, a complication of the gearing adjustment process, increased noise during operation, and a decrease in the mechanical efficiency of the transmission. Gear drives are most widespread on marine four-stroke trunk diesel engines (Fig. 4.27 *a*, *b*). The relatively small interaxal distances of such engines make the gear transmission quite compact.

In two-stroke LSEs, gear drives, due to the large dimensions of the gears, are more complex and cumbersome (Fig. 4.23 *a*). However, when choosing, their high reliability is sometimes the main determining factor. Thus, the Wärtsilä company, which inherited the developments of the Sulzer company, as well as the Mitsubishi company, prefer this type of transmission for their engines. In order not to complicate the design of the drive, the camshafts of modern LSEs are located at the middle level, and forces are transmitted to the valve through a transmission mechanism, usually hydraulic.

In some designs of HSEs with overhead camshafts, helical gears are used, which make it possible to reduce the dimensions of the drive and the number of idler links (Fig. 4.23 *f*, 4.27 *d*). It is interesting to note that it was this type of transmission that was used by R. Diesel in the first diesel engine he designed (Fig. 1.2).

Gears of cylindrical gears are made with spur or helical gears. Bevel gears are usually made with spur gears. To ensure correct installation of the gears during assembly, appropriate marks are applied to them. A set of transmission gears is placed in a closed box, which communicates with the engine crankcase, and is provided with constant lubrication entering the contact area of the gears through special injectors.

Chain drives are usually used at large distances between the axes of the camshaft and crankshaft, with a middle or upper camshaft position (Fig. 4.27 *e*, *g*). These transmissions are distinguished by their relative simplicity of design, compactness and relatively low noise. In addition, the chain drive makes it easy to compensate for changes in the distance between the shafts when the diesel core is heated and to drive auxiliary mechanisms, such as balancers, air distributors, fuel pumps, etc., with relative ease. Traditionally, the MAN company uses chain drives in its designs (Fig. 4.27 *g*).

The most significant disadvantage of chain drives is that during operation, wear and stretching of the chain occurs, as a result of which vibrations increase, operating noise increases, and valve timing changes. To maintain the chain in tension, lever tensioners are usually used, which are additional sprockets mounted on a lever suspension (Fig. 4.27 *e*, *g*, *h*). The tensioner sprocket is pressed against the chain using a spring, the pre-tension of which is adjusted using a screw or hydraulic mechanism. To prevent lateral vibrations, long sections of the chain move along guide plates called *dampers*, which are lined with rubber or other polymer materials on the working side.

In marine diesel engines, bushing-roller chains are usually used (Fig. 4.27 *f*) assembled from external and internal links. The outer link consists of two plates and two fingers pressed into them,

For preliminary review

and the inner link consists of two plates, two bushings and two rollers. The bushings are pressed into plates, and rollers are placed on them with a gap to ensure free rotation. Pins, bushings and rollers form bearing units that reduce friction, both in the chain itself and when interacting with the teeth of the sprockets. The roller, when in contact with a tooth, rolls along it, reducing friction losses and reducing tooth wear. The link plates serve as connecting elements and work in tension. The distance between the pin axes is called *chain pitch*. Together with the roller diameter, the chain pitch determines the main geometric characteristics of the sprocket tooth profile.

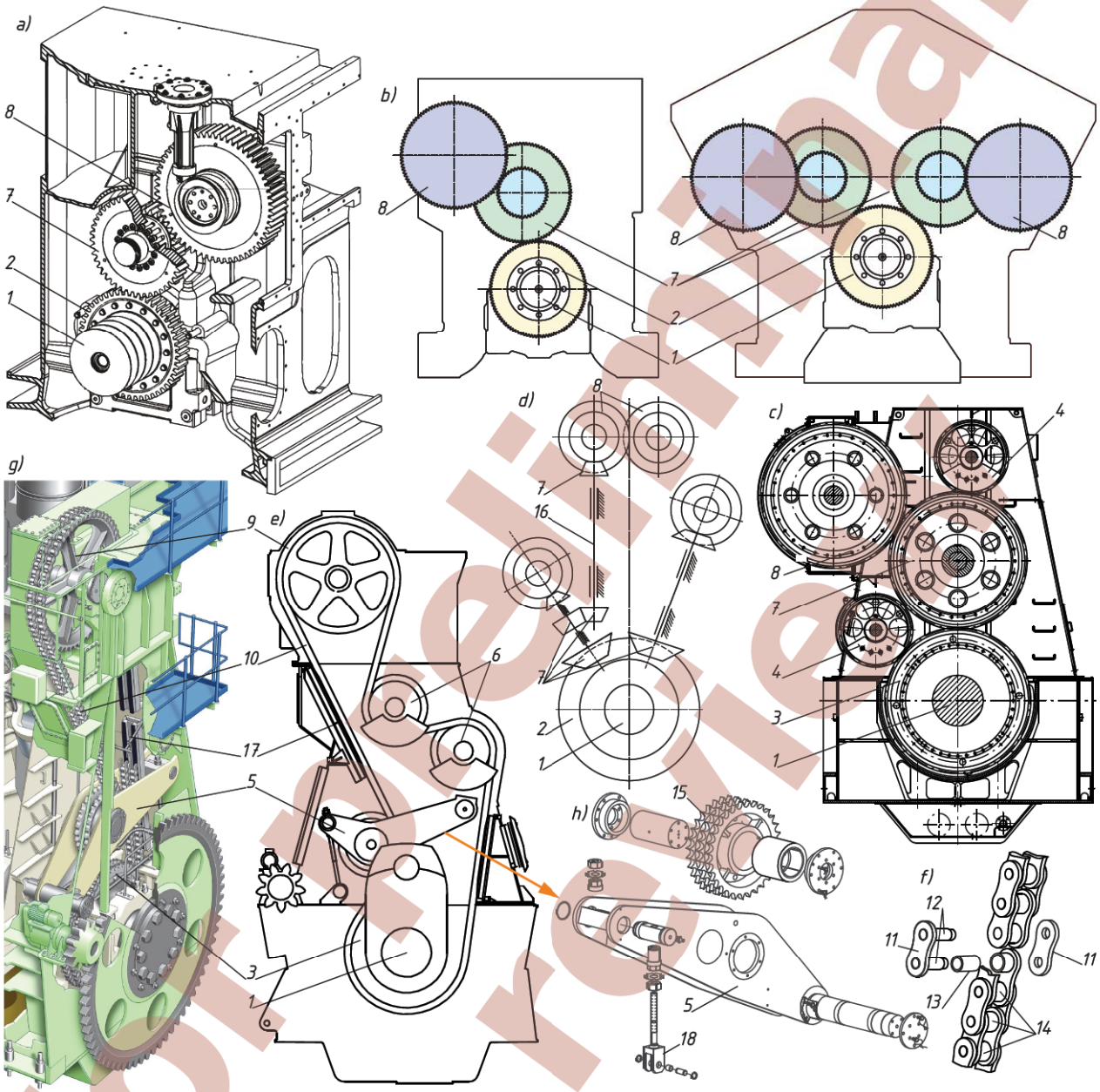


Figure 4.27 – Camshaft drives and their elements: *a* – four-stroke diesel engine MaK 43C; *b* – gear arrangement for in-line and V-shaped four-stroke engines; *c* – gear drive LSE series RTA from Wärtsilä; *d* – with bevel gears; *g*, *e* – chain drive LSE series K, L, S from MAN; *f* – fragment of a bushing-roller chain; *h* – chain tensioner: 1 – crankshaft; 2, 3 – drive sprocket (gear); 4, 6 – gears (sprockets) of the balancer drive; 5 – chain tensioner lever; 7 – idler gear; 8, 9 – gears (sprockets) of the camshaft drive; 10 – chain; 11 – side pads; 12 – fingers; 13 – bushings; 14 – rollers; 15 – tensioner sprocket; 16 – idler shaft; 17 – chain stabilizer; 18 – chain tensioner (adapted from [11, 13, 19, 23])

If it is necessary to transfer large forces from the crankshaft to the camshaft, chains made in several rows, from two to four, are used.

The axles of all sprockets and chains are lubricated with oil using injectors.

Preliminary version

The transmission mechanism serves to transmit force from the camshaft to the valves. In marine MSEs and HSEs, mechanical transmission mechanisms are most widely used. The kinematic diagram of the transmission mechanism largely depends on the selected engine layout and, first of all, on the location of the camshaft. Engines with an overhead camshaft are characterized by the simplest kinematic schemes, when the cam acts on the valve through a single (Fig. 4.26 *e*) or double-arm lever, as well as directly on the upper valve plate, which acts as a pusher (Fig. 4.26 *f*). The minimum number of parts makes it possible to reduce the mass of moving parts, reduce the inertial forces acting in the drive, and reduce the dimensions of the engine. For this reason, this scheme is typical mainly for water supply systems.

Most shipboard MSEs and HSEs are characterized by the transmission mechanism arrangement shown in Fig. 4.15.

The main disadvantage of a mechanical valve drive is the presence of gaps to compensate for thermal expansion of its parts, which prevent the possibility of incomplete closing of the valve. Due to the presence of gaps, the movement of the valve begins and ends with an impact, causing additional stress in the seat and disc, spring and on the contact surfaces of the transmission mechanism. The size of the gaps largely depends on the size of the drive parts and their number; the longer the parts and the more of them, the more they expand and the larger the thermal gap must be provided. In trunk engines, where the distance between the camshaft axis and the rocker arm axis is relatively small, the simplicity and reliability of the mechanical drive are decisive factors when choosing the type of transmission mechanism. To eliminate the above drawback, some HSEs and MSEs use devices that allow the clearances in the mechanical drive to be reduced to zero during engine operation without the risk of incomplete closing of the valves.

For LSEs with a large height, the mechanical drive parts can be quite large, which requires significant thermal clearances. In addition, a large mass of parts leads to an increase in inertia forces in the drive. Repeated attempts to get rid of gaps and associated shock loads in LSE drive mechanisms led to a significant complication of the design. This was one of the main reasons why all manufacturers of engines of this class switched to hydraulic transmission mechanisms.

The hydraulic valve drive is used in some MSEs (Fig. 4.15, 4.19).

The elements of the mechanical transmission mechanism are shown in Fig. 4.28. Next, their design and operating conditions will be discussed in more detail.

The pushers receive the force from the cam washers and transmit its axial component to other elements of the transmission mechanism, and lateral forces to the guide surfaces of the frame. To reduce the inertia forces of the moving parts of the transmission mechanism, the drive parts must have as little mass as possible. For these purposes in HSEs use pushers with a cylindrical guide surface in the form of hollow cups (Fig. 4.28 *a*) or T-shaped (Fig. 4.25 *b*) pushers, with a flat or spherical working surface.

To ensure more uniform wear of the guide and working surface, as well as to reduce friction forces, the pushers are given a rotational movement. To do this, the axis of the pusher with a flat working surface is shifted by 1.5...3 mm relative to the cam (Fig. 4.28 *d*), and with a spherical working surface the cam is made slightly conical (Fig. 4.28 *e*). Due to this, when the working surfaces of the cam and the pusher interact, a torque arises that rotates the pusher housing at a certain angle.

To ensure shock-free operation and reduce noise, some water pumps use pushers with a hydraulic thermal gap compensator, the general layout of which is shown in Fig. 4.28 *c.* The housing of such a pusher houses a hollow, tightly fitted piston, the upper part of which is closed by a plug with a hole for the heel of the pusher rod, and a non-return plate or ball valve is installed in the lower part. The closed internal cavity of the piston communicates through a system of grooves and holes in the housing of the housing and the piston itself with the engine oil line. Between the bottom of the pusher housing and the lower part of the piston, a closed cavity is formed in which a spring is installed, under the action of which the piston tends to occupy the upper position limited by the stop in the housing. The spring acting on the piston is several times weaker than the valve springs, so it is able to select gaps in the transmission mechanism, but is not able to open the valve. In the ab-

For preliminary review

sence of force from the cam, the piston moves upward under the action of the spring until it removes all the gaps in the drive mechanism. In this case, oil from the piston cavity fills the sub-piston cavity through the open valve. When the cam acts on the pusher housing, the pressure in the sub-piston cavity increases sharply, which leads to the closure of the check valve. The oil locked in the sub-piston cavity prevents the reverse downward movement of the piston, as a result of which it moves along with the housing, transmitting force to the pusher rod. As the pusher moves, some of the oil is squeezed out through the gap, but in the absence of force from the cam, oil losses are compensated by its flow through the valve. A change in the temperature regime of the engine leads to a change in the lengths of the drive elements, which leads to a change in the amount of oil in the sub-piston cavity of the pusher, providing a shock-free elastic connection between the pusher housing and the piston.

In for MSE and some HSEs, cylindrical pushers have become widespread, in the lower part of which a roller is installed on the axis, rolling along the surface of the cam (Fig. 4.28 *f*). Roller tappets can significantly reduce friction losses, and in tandem with them, cams with a concave profile can be used, which ensures more efficient operation of the valve mechanism with a relatively smooth lift-off and landing of the valves on the seats. At the same time, an increase in the mass of the pusher leads to an increase in inertial forces, which leads to an increase in loads on the parts of the transmission mechanism, sharply increasing the noise of its operation.

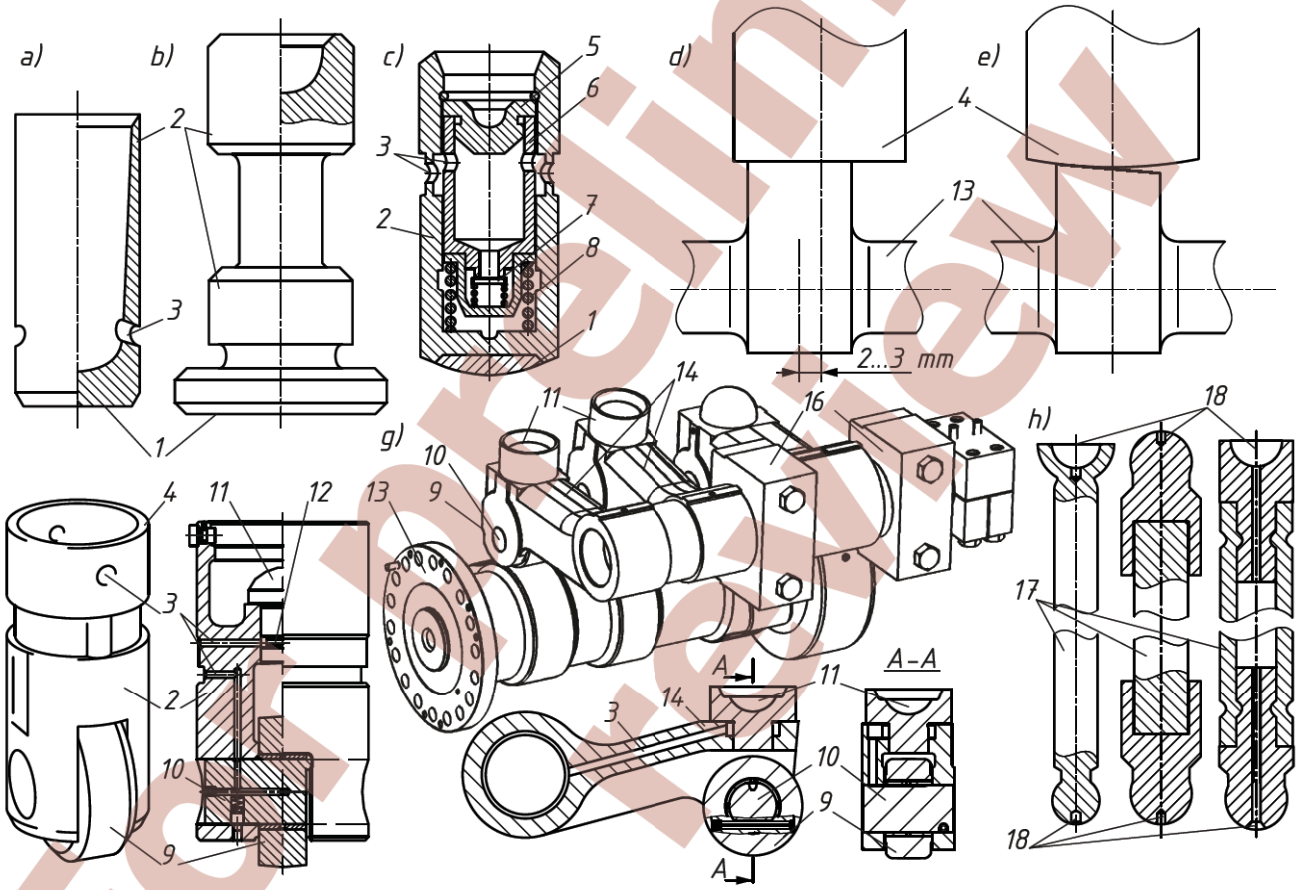


Figure 4.28 – Pushers and pusher rods: *a* – cylindrical pusher; *b* – T-shaped pusher; *c* – cylindrical pusher with hydraulic thermal gap compensator; *d*, *e* – diagrams for installing pushers for turning them around the axis; *f* – cylindrical roller pusher; *g* – lever roller pusher; *h* – some types of pusher rod tips; *i* – pusher rod assembly: 1 – working surface; 2 – guide surface; 3 – holes for oil supply; 4 – pusher housing; 5 – hydraulic piston cover with a hole for the pusher rod; 6 – hydraulic piston; 7 – check valve; 8 – hydraulic piston spring; 9 – roller; 10 – roller axis; 11 – drummer; 12 – guide pin; 13 – camshaft; 14 – lever; 15 – lever axis; 16 – fastening of the lever axis; 17 – pusher rod; 18 – tip (adapted from [14])

In some HSEs, the roller is mounted on a needle bearing, and in MSEs in which more significant forces are applied in the drive, preference is given to plain bearings in the form of a solid bushing

For preliminary review

made of antifriction material. The roller pushers are secured against axial rotation using special pins installed in the frame, which fit into vertical slots on the pusher housing.

Lever swinging pushers that make angular movements relative to the hinge support consist of levers with strikers and rollers resting on the camshaft cams (Fig. 4.28 *g*). The pushers are mounted on a common or separate axis installed parallel to the camshaft axis. In terms of layout, such mechanisms turn out to be more cumbersome, however, the ease of dismantling and installing the pushers along with the axles through the hatches of the camshaft compartment has contributed to the spread of this type of pushers on some modern MSEs and HSEs.

The guide surfaces of cylindrical pushers, roller axes and levers are lubricated under pressure. The cams and working surfaces of the pushers are lubricated with oil mist or oil supplied through special injectors.

Push rods serve to transfer forces from the pusher to the rocker arm. Structurally, the rods must be sufficiently rigid and at the same time have a minimum mass. They are usually made in the form of solid rods or hollow tubes, at the ends of which steel tips made of high-hard steel are pressed. The ends of the tips are given a concave or convex spherical shape, which follows the shape of the holes in the pusher and on the rocker arm (Fig. 4.28 *h*, *i*). The supporting surfaces of the tips are heat treated, ground and polished. For rods made of steel rods, the ends can be made integral with the rod.

On engines, the frame and cylinder covers of which are made of aluminium alloys, in order to proportionally elongate the push rods during the heating process, their rods are also made of aluminium alloys.

In a number of designs, the internal cavity of the pusher rod is used to supply oil under pressure to the parts of the gas distribution mechanism located in the valve box. In this case, axial drillings of small diameter are provided in the tips of the rods.

To protect the rods from external influences and damage, they are enclosed in special channels made as vertical wells in the engine frame or in the form of separate insert tubes, which are sealed with rubber rings at the junction with the frame and the cover. Quite often use these channels to drain oil from the valve box of the cylinder cover into the engine crankcase.

Valve drive rocker arms serve to transmit force from the rod to the valve. They are double-armed levers, which are made of a T-section or I-section to increase rigidity. To reduce inertial loads in the transmission mechanism, usually the rocker arm facing the valve is made longer than the arm on the rod side, thus, the required valve lift height is ensured with significantly less movement of the pushers and rods.

To simultaneously drive two valves, a scheme can be used in which two levers are used, main and additional (Fig. 4.29 *a*). The main lever acts on one of the valves and on an additional lever from which the second valve is driven. In a number of engines, the rocker arm is made in the form of a fork with two arms, each of which acts on its own valve (Fig. 4.29 *b*, *e*). In addition, the rocker arm can act not directly on the valves, but through special traverses mounted on cylindrical guides (Fig. 4.29 *c*). In the first two cases, the rocker arm, interacting with the end of the valve stem in the contact zone, rolls with sliding, which creates lateral forces causing increased wear of the valve guides.

In the second case, lateral forces are perceived by the guide traverse, and only the axial force acts on the valve. This allows you to increase the service life of guide bushings and valves. It is for this reason that the latter design is increasingly used in MSE and some HSEs.

If the engine is not equipped with hydraulic thermal gap compensators, then a threaded hole is provided on one of the lever arms for an adjusting bolt. The head of the bolt is given a concave or convex spherical shape, which follows the shape of the tip of the pusher rod or hole on a special thrust bearing, through which the head acts on the valve stem or traverse. The bolt has a central channel and a groove for supplying oil to the tip of the rod. On the opposite side, a lock nut is screwed onto the adjusting bolt, and the protruding part ends with an external or internal hexagon or a slot for a screwdriver for turning the bolt while adjusting the thermal gap. If there is a traverse, for alignment relative to the guide, it is also equipped with an adjusting bolt.

For preliminary review

The end of the lever directly acting on the valve or traverse, if it is not equipped with an adjusting screw, is made in the shape of a cylinder.

Most often, the levers are hinged on special axes. To do this, there is a boss in the lever housing, in which a cylindrical hole is bored, equipped with a bearing sleeve made of anti-friction material. The axles with rocker arms are mounted directly in the cylinder cover housing or on special racks (Fig. 4.29 *d*), which are fixed in a given position using pins and bolted to the cylinder covers. Rocker arms are usually kept from longitudinal movement on the axis by spiral springs, spring washers, etc.

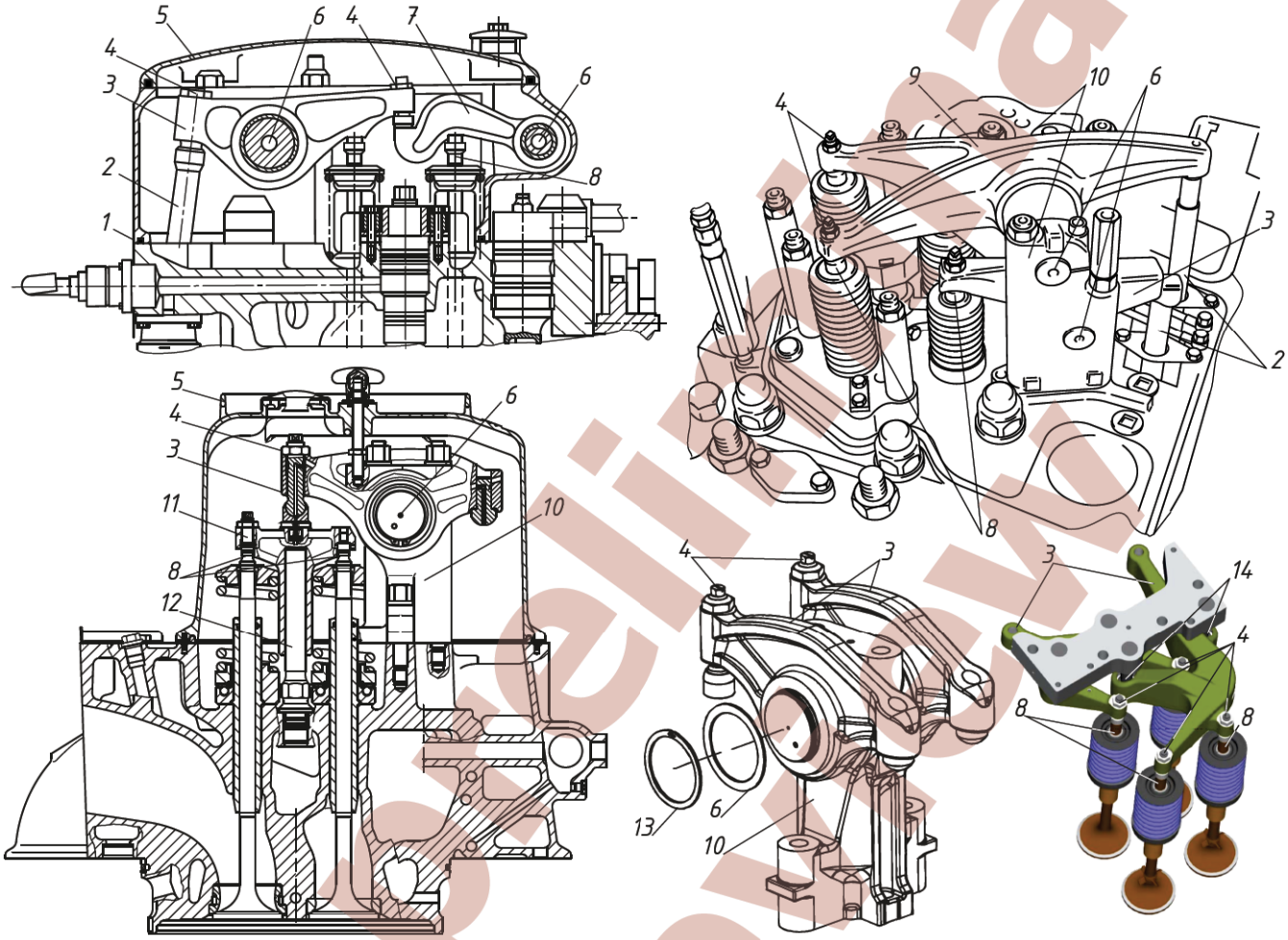


Figure 4.29 – Layout diagrams of valve drive mechanisms: *a* – four-lever Sulzer ZL 40 engine; *b* – with a fork rocker of a TBD 645 diesel engine, from Deutz MWM; *c* – with valve drive through the traverse of the MaK 25C diesel engine; *d* – stand with the axis of the rocker arms of the MaK 43C engine; *e* – suspension of levers on the hinge pins of the MAN L58/64 diesel engine: 1 – valve box; 2 – push rod; 3 – rocker arm; 4 – adjusting bolt; 5 – valve box cover; 6 – rocker axis; 7 – additional lever; 8 – valve stems; 9 – fork lever; 10 – rocker arm axis stand; 11 – valve drive traverse; 12 – valve drive traverse guide; 13 – spring lock washer; 14 – pin with ball joint (adapted from [7, 13, 14, 22, 24])

Since in the hinged connection of the rocker arm and the axle only the lower part of the bushing is loaded, a number of designs abandoned its use, replacing the cylindrical hinges with spherical stops, onto which the lever is placed with special recesses that repeat the shape of the stop (Fig. 4.29 *e*). This design made it possible to reduce the weight of the lever and distribute the loads more evenly over the hinge area. In addition, the lever has acquired the ability to self-align, leveling out assembly inaccuracies and distortions resulting from uneven wear of valve components and the drive mechanism.

Rocker arms are cast from cast iron or steel using precision casting followed by machining of all working surfaces.

For preliminary review

Gas distribution ports are divided into exhaust and purge ports. They are made in the form of rectangles, parallelograms, trapezoids, as well as oval or round. The number and location of ports depends on the chosen gas exchange scheme and the diameter of the cylinder.

Modern LSEs use a direct-flow valve purge scheme, which assumes the presence of only purge ports, which are located along the entire circumference of the cylinder in its lower part (Fig. 4.30 *a*). When choosing the shape and size of ports, they strive for their larger total width, reducing the thickness of the lintels between them, as far as strength conditions allow. This makes it possible to significantly reduce the height of the ports, thereby reducing the share of lost working stroke on the gas exchange organs. The total width of the ports for a direct-flow purge scheme is usually 0.60...0.75 of the cylinder perimeter, and the height is 0.05...0.09 of the piston stroke. The largest flow area is provided by rectangular ports (Fig. 4.30 *b*), but stress is concentrated at their corners. For this reason, the upper and lower edges of the ports are made rounded, with a rounding diameter equal to the width of the port (Fig. 4.30 *c*), and in a number of designs, ports are made as several intersecting drillings (Fig. 4.30 *d*). To reduce the resistance to the incoming air flow, the inlet edges of the purge ports are slightly rounded.

In relation to the plane perpendicular to the cylinder axis, the purge ports are positioned tangentially (Fig. 3.9 *e*) at an angle of 15...25° or radially.

The tangential inclination to the cylinder axis helps to create turbulence of the fresh charge and improve the mixture formation process, however, at significant air velocities at the entrance to the cylinder, the bulk of it can be pressed by centrifugal forces against the cylinder walls. In this case, a so-called dead zone is formed along the axis, with inactive contents.

Radially directed ports have the simplest design and technology and manufacturing, however, along the jumpers, wall dead zones can also form here, which is associated with the uneven distribution of fresh charge along the perimeter of the cylinder. To eliminate this drawback, purge ports are sometimes used, the vertical axis of which is deviated from the cylinder axis by an angle of 7...10° (Fig. 4.30 *e*). Thus, the upward flow is evenly distributed over the entire surface of the cylinder liner.

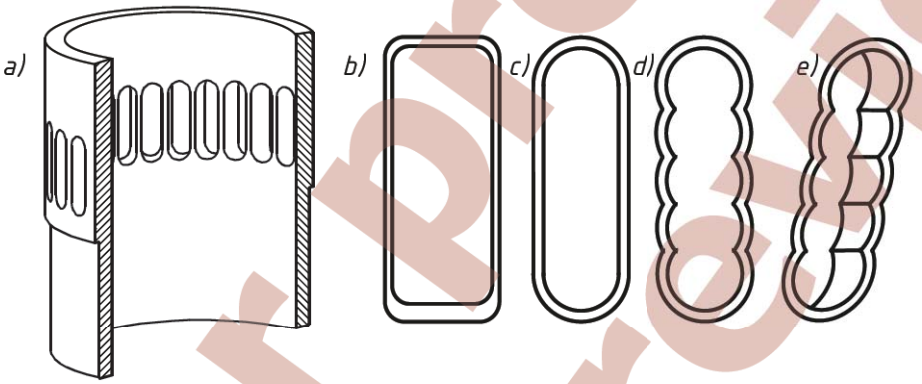


Figure 4.30 – Gas distribution ports of two-stroke diesel engines with direct-flow purge: *a* – location of ports on the cylinder liner; *b* – rectangular; *c* – with rounded ends; *d* – segmental; *e* – segmental with a tilt of the vertical axis

Two-stroke engines with direct-flow slot scavenging have, in addition to scavenging ports, exhaust ports. This scheme is used in diesel engines with oppositely moving pistons, in which air enters through purge ports, and exhaust gases exit through exhaust ports located along the entire circumference in the opposite part of the cylinder. Purge ports, as a rule, have a tangential location, and outlet ports have a radial location.

4.5.7 Hydrostatic transmission mechanisms

Hydraulic transmission mechanisms have become prevalent in LSE and some MSE. Their main advantage is the ability to transmit large forces with relatively small dimensions and masses of structural elements. This makes it possible to reduce the inertial forces acting in the drive, increase its reliability, and reduce operating noise.

For preliminary review

The hydraulic drive consists of a piston pump-actuator driven by a cam washer, a hydraulic transmission and an actuator that acts on the valve stem at the moment of its opening.

The valve is closed with springs (Fig. 4.21 *a*) or with the help of an air piston attached to the valve stem (Fig. 4.20). The piston, in turn, is placed in a cylinder, to which compressed air is supplied from the ship's air system.

The actuator pump is a piston pump driven by the camshaft cam washer. For a compact design, the pump-actuators are assembled in the same housing with fuel pumps and indicator drives. A general view of the pump block and a cross-section of the pump-actuator of the modern two-stroke low-speed diesel engine RTA 72 U from Wärtsilä is shown in Fig. 4.31.

The hydraulic piston of the actuator is driven by a roller tappet, which moves upward under the action of a cam washer, and downwards under the action of a spiral spring, which is fixed between the tappet and the hydraulic cylinder. Axial rotation of the pusher is prevented by a guide pin. The hydraulic piston is inserted into the cylinder and rests on the heel of the pusher. It is equipped with a system of seals installed in annular grooves on its cylindrical generatrix.

The valve actuating oil is usually supplied under pressure from the camshaft lubrication system through a non-return valve at the top of the actuator hydraulic cylinder. Oil leaks through the piston seals, as well as from under the pressure limitation valve, flow into the pump block housing.

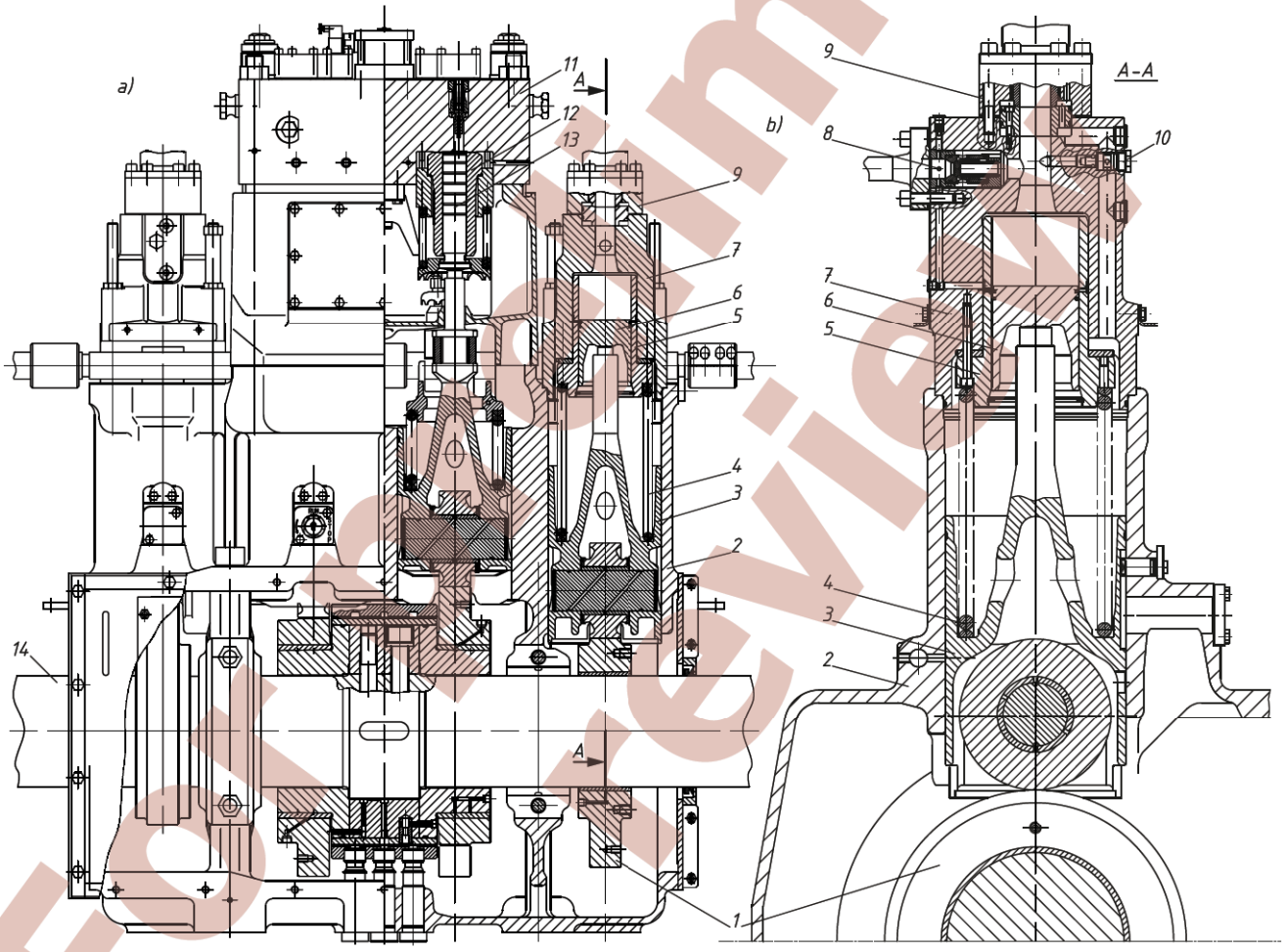


Figure 4.31 – Block of fuel supply pumps (*a*) and actuators for the hydraulic drive of the exhaust valves (*b*) of the RTA 72 U diesel engine from Wärtsilä: 1 – camshaft cam; 2 – pusher housing; 3 – roller pusher; 4 – return spring; 5 – hydraulic cylinder; 6 – hydraulic piston; 7 – actuator housing; 8 – non-return valve of the make-up channel; 9 – flange of the high pressure pipe of the valve drive; 10 – pressure limiting valve; 11 – fuel pump roof; 12 – fuel pump plunger bushing; 13 – fuel pump plunger; 14 – camshaft (adapted from [11])

The pump-actuator is connected to the actuator of the exhaust valve drive by a high-pressure

For preliminary review

line, which, depending on the size of the engine, can be continuous or consist of several pipelines connected to each other and other drive elements using flanges attached to the tubes (Fig. 4.31).

The high-pressure tubes are covered on top with a protective braid, which is a thin-walled solid pipe or woven fabric made of steel wire. This allows, in the event of a pipe break, to disperse the oil stream, avoiding injury and oil getting on heated engine parts.

The exhaust valve with the drive mechanism is included in the valve assembly, which is mounted on the cylinder cover (Fig. 4.17 *a*).

The valve assembly of all modern LSEs has a replaceable seat in the lower part of the housing. Together with a replaceable valve guide, this allows you to replace worn elements without replacing the most expensive housing parts. The housing of the valve assembly is cooled by water, which comes from the cylinder cover through the inlet pipe, and drains from the top of the housing through the outlet pipe (Fig. 4.20). To stabilize the thermal conditions of the valve assembly, a throttle washer is installed in the outlet pipe to limit the amount of cooling water passing through the outlet valve housing. To monitor the condition of the cooling cavity and its cleaning, there are removable hatches on the housing.

The actuator consists of two pistons, hydraulic and air. The hydraulic cylinder is installed above the air cylinder in the upper part of the valve assembly housing (Fig. 4.20, 4.32).

The hydraulic piston is used to open the exhaust valve by pressing the end of its rod. It has two metal piston rings and a damping device designed to reduce the speed of the valve as it seats. This significantly reduces shock loads in the actuator and extends the life of the valve itself and its seat.

The damping device of the hydraulic piston LSE of the RTA series from Wärtsilä is a belt protruding inside the cylinder, which, when the piston approaches the upper position, enters a cylindrical recess on its housing, forming a closed volume (Fig. 4.20). The oil, once locked in this volume, comes out of it through calibrated holes or gaps between the walls. As a result, the valve seating speed is sharply reduced.

In engines of the K, L, S series from MAN, the damper device is made in the form of a spring-loaded piston protruding from the hydraulic piston housing (Fig. 4.33). Before closing the valve, the damping piston enters a cylindrical recess on the hydraulic cylinder housing, preventing the free exit of oil through the supply channel. As a result, a closed volume is formed, from which the oil slowly leaves through radial drillings in the upper head of the hydraulic piston and then through a calibrated hole in the bottom of the damper piston.

A rod is installed in the upper part of the hydraulic cylinder to control the lifting and rotation of the valve when the engine is running (Fig. 4.33), as well as a throttle valve designed for automatic deaeration of the oil system. When the exhaust valve is closed, the pressure in the hydraulic cylinder decreases and the spring opens the deaeration valve slightly. When the pressure in the cylinder begins to increase, part of the oil, along with the air, passes through this valve and is drained through a special channel into the cavity around the air cylinder from where, along with leaks from under the piston, it is discharged into the leak collection system. With a further increase in pressure, the deaeration valve closes and does not affect the further operation of the hydraulic drive.

The air piston serves to close the exhaust valve and is connected to its rod using conical «crackers». The piston is sealed relative to the air cylinder using solid rings made of polymer materials, which are installed in the annular grooves on the piston housing.

The air cylinder is mounted directly on the valve housing below the hydraulic cylinder. To prevent air leaks, the valve stem at the base of the cylinder is sealed with solid polymer rings. Rod seal wear is determined by using a drain hole located in the space between the O-rings.

To protect against excessive pressure in the air cavity of the valve drive, a pressure limiting valve is installed on the valve assembly housing (Fig. 4.33).

In a number of designs, an additional cavity of sealing air is located under the bottom of the air cylinder (Fig. 4.33). Air is supplied into this cavity below the sealing rings of the valve stem through a calibrated hole. This design solution prevents the penetration of exhaust gases and solid particles into the cavity of the pneumatic cylinder, which reduces wear on the working surfaces and contamination of the elements of the pneumatic valve actuator system.

For preliminary review

4.5.8 Hydrostatic valve drive with electronic control

The hydraulic drive of exhaust valve control actuators in combination with hydraulic transmission mechanisms has become widespread on new engines of the ME series from MAN, RT-flex from Wärtsilä and UEC Eco-Engine from Mitsubishi (Fig. 4.34).

On all engines, the use of a hydraulic drive of actuators is an element of an integrated control system for operating processes using microprocessor technology, which, in addition to gas distribution, includes control of fuel supply, lubricator pumps, and starting air.

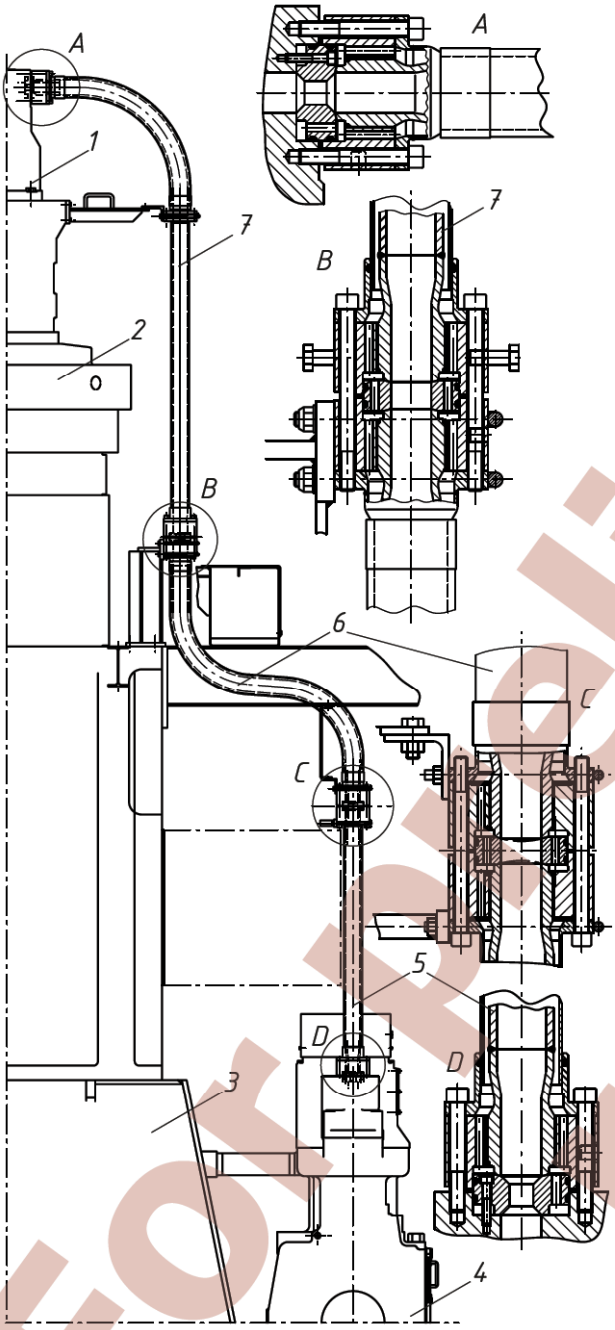


Figure 4.32 – High pressure line for the hydraulic valve drive of low-speed engines of the RTA series from Wärtsilä: 1 – valve drive; 2 – cylinder cover; 3 – engine frame; 4 – pump block; 5 – lower pipe with flanges; 6 – idler elbow; 7 – upper knee (adapted from [11])

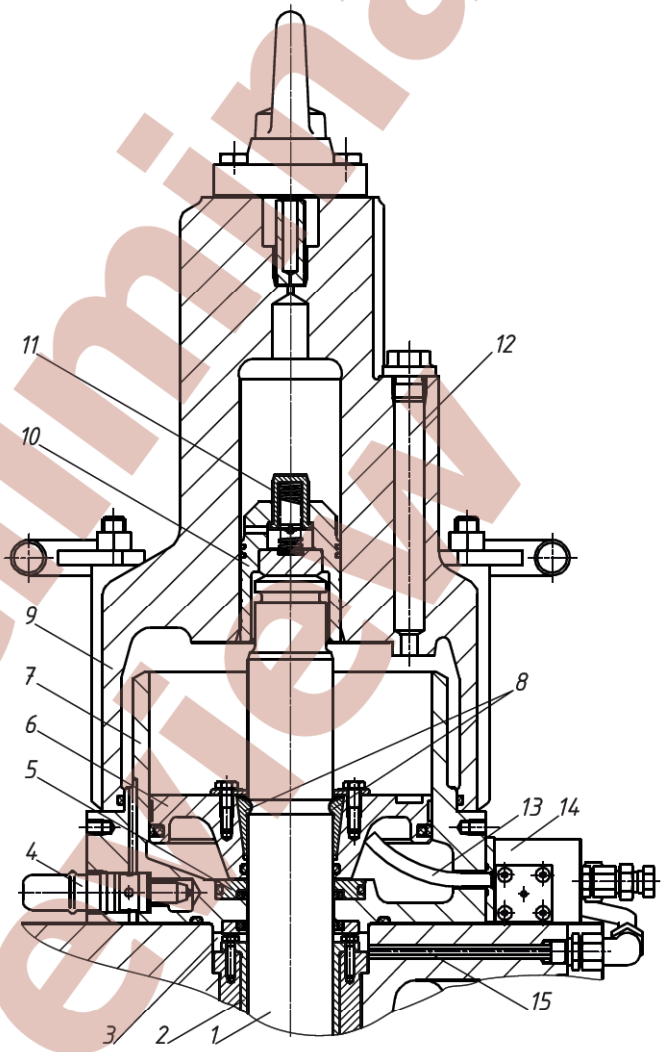


Figure 4.33 – Hydropneumatic drive mechanism for LSE valves of the K, L, S series from MAN: 1 – valve stem; 2 – valve guide bushing; 3 – sealing air cavity; 4 – pressure limitation valve; 5 – valve stem seal; 6 – air piston; 7 – air cylinder; 8 – conical crackers for fastening the air piston; 9 – hydraulic cylinder housing; 10 – hydraulic piston; 11 – piston of the damper device; 12 – channel for installing a valve lift control device; 13 – air supply pipe; 14 – valve block; 15 – sealing air supply line (adapted from [22])

The hydraulic drive makes it possible to significantly simplify the design of the mechanical part

For preliminary review

of the engine; in particular, there is no need for camshafts and their drive. In addition, it makes it possible to quite easily change the operating modes of individual elements of engine systems. In particular, in relation to the gas distribution system, the hydraulic drive allows, depending on the operating mode, to change the valve timing and the law of valve movement.

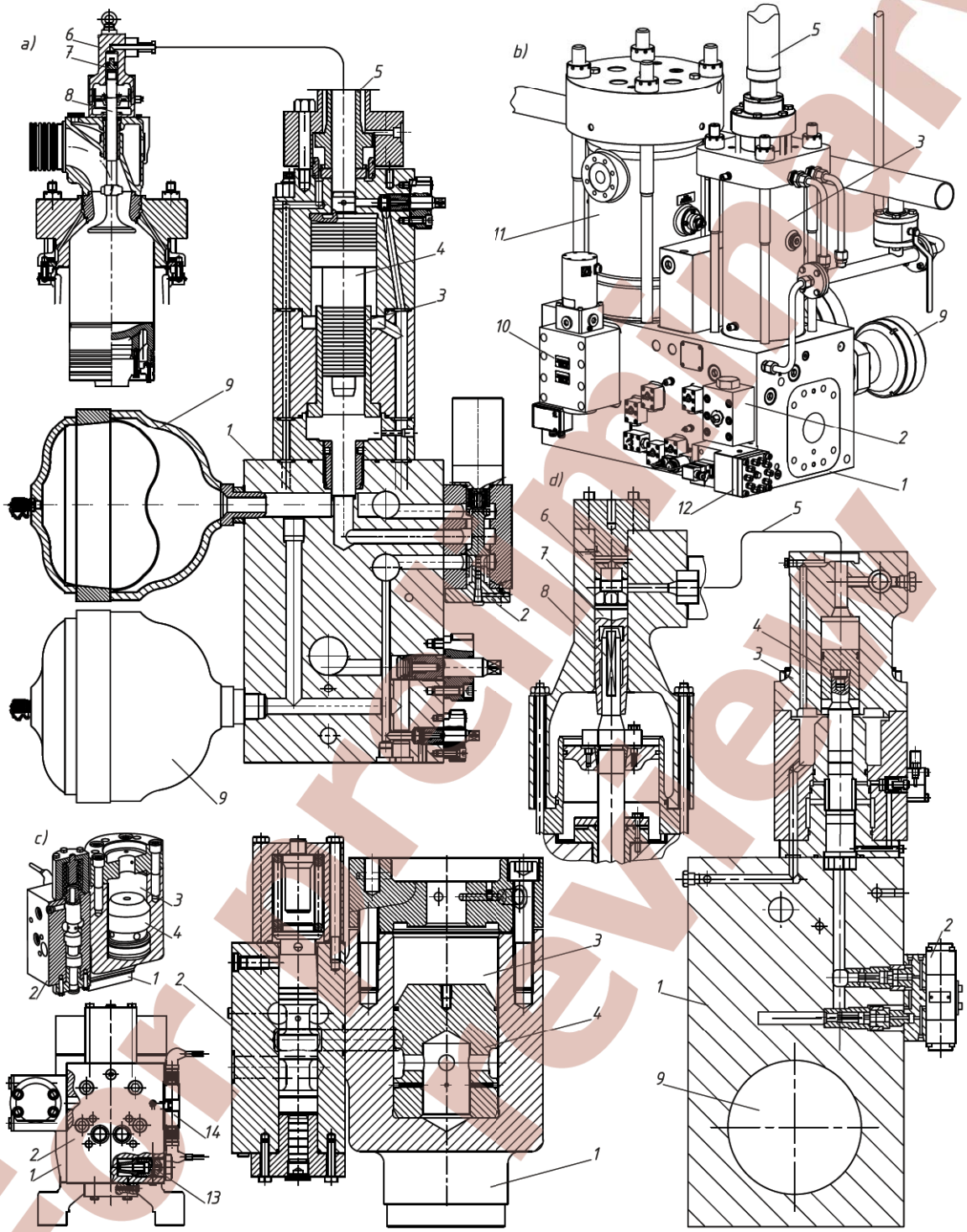


Figure 4.34 – Hydraulic drive of gas distribution mechanism actuators: *a* – engines ME series K, L, S from MAN; *b* – general view of the hydraulic block of MAN ME engines from the valve drive actuator side; *c* – general view and longitudinal section of the hydraulic actuator for diesel engines of the RT-flex series from Wärtsilä; *d* – engines of the UEC Eco-Engine series from Mitsubishi: 1 – hydraulic unit housing; 2 – control spool for the exhaust valve actuator; 3 – hydraulic cylinder; 4 – actuator piston; 5 – high pressure hydraulic line; 6 – hydraulic cylinder of the valve drive; 7 – valve drive piston; 8 – exhaust valve stem; 9 – pressure accumulator; 10 – spool control of the fuel pump actuator; 11 – fuel pump actuator; 12 – lubricator block; 13 – control oil filter; 14 – block of electrically controlled valves (adapted from [25, 26, 27, 28])

For preliminary review

To implement this concept, engines are equipped with actuators in which the valve drive piston moves upward under the influence of oil pressure from the control line entering the sub-piston space. The oil supply is controlled through a system of spools, which, in turn, are controlled using solenoid valves that receive a signal from a microprocessor control unit. Oil is pumped into the control line using several pumps, some of which are driven by the engine itself and provide consumers with oil during its operation, and some are electrically driven, providing oil supply during diesel start-up and low rotation speeds. The pressure in the control line, for example in engines of the RT-flex series from Wärtsilä, is maintained at 20 MPa.

To maintain constant pressure in the sub-piston cavity of the actuator, a number of designs use gas pressure accumulators (Fig. 4.34 *a*, *b*), which are a metal cylinder divided into two cavities using an elastic diaphragm. Gas (most often nitrogen) is pumped into one cavity under high pressure, and the other cavity is filled with oil. In the absence of a control signal, oil from the control line enters the accumulator cavity, filling it and compressing the gas through the diaphragm. The accumulator is charging. When the spool opens, oil from the accumulator under pressure created by the gas spring enters the sub-piston cavity of the actuator, activating the force transmission mechanism to the exhaust valve.

In the design developed by Mitsubishi and Wärtsilä, the role of the accumulator is played by a pressure line, which has a fairly large volume. In the Mitsubishi design, the accumulator cavity is made directly in the housing of the hydraulic unit (Fig. 4.34 *d*). In the Wärtsilä design, the actuator is made in the form of a separate block installed directly on the control line, which is a thick-walled pipe running along the upper platform of the engine (Fig. 4.34 *c*).

To increase the speed of the hydraulic drive, a two-stage spool control system is used. The first stage with electromagnetic control must have relatively small dimensions, since increasing the size of the solenoid core increases its hysteresis and, accordingly, the time for its magnetization. Therefore, a relatively small electromagnetic drive controls the main spool, which in turn supplies oil to the main spool control cavity.

The successful experience of introducing electronically controlled engines of the ME series pushed MAN specialists to further develop this concept in a more compact design. Work on the creation of new types of drive mechanisms for gas distribution valves and fuel supply systems was carried out in the period from 2012 to 2014. Bench testing of the laboratory 4S50ME-T9 engine was carried out in 2015, and testing on three cylinders of the commercial 8G95ME engine began the following year.

Subsequently, these developments formed the basis of the new engine platform of the G95ME series – Mk 10.5. The further desire of MAN engineers to reduce the weight of the new generation engines led to the development of direct drive valve control components located directly on the exhaust valve housing. This made it possible to reduce their weight compared to the ME-C series of the Mk 9 platform by approximately 30%. The design of the exhaust system for the new generation engines has not undergone significant changes, but the design of the valve drive, as well as the fuel injection system, has been significantly redesigned. The new exhaust valve control technology is called Top-Controlled Exhaust Valve (TCEV).

The general structure of the TCEV mechanism is shown in Fig. 4.35.

The abbreviation TCEV is used for the entire exhaust valve assembly in which a piston in a control sleeve within the actuator directly actuates the valve spindle.

The actuator unit is mounted directly on the exhaust valve housing. The main components of the drive system are the sleeve of the working hydraulic cylinder (item 8, Fig. 4.35) and piston 9, which acts as a spool, which with its end directly acts on the upper part of the exhaust valve rod 1.

The supply of oil to the hydraulic cylinder is controlled by a hydraulic distribution block mounted on the designated surface of the TCEV actuator unit.

Control oil flows are controlled by a two-stage electro-hydraulic system, which is similar to that used on ME series engines. To do this, a platform with two control valves mounted on it is installed on the drive housing 7, one of which controls fuel injection (Electronic Fuel Injection – ELFI), and the other controls the exhaust valve (Proportional Exhaust Valve Actuation – PEVA).

For preliminary review

To stabilize the control oil pressure, membrane-type gas accumulators 14 are installed on the platform. The outlet valve is opened by supplying hydraulic oil under pressure into the oil cylinder of the TCEV actuator. Closing occurs under the action of a pneumatic spring, consisting of a cylinder 4 and an air piston 5. The operating procedure of the TCEV drive is shown in Fig. 4.36.

The diagram of the hydraulic drive of the actuator of the RT-flex series engines, and the order of its operation are shown in Fig. 4.37.

The spool piston of the exhaust valve spindle drive has two working end surfaces on the upper face of the spool and on the end surface of the first stage (Fig. 4.36 *e*, pos. 2 and 3). Together with the bushing, these surfaces form two hydraulic cavities, the main and additional ones, respectively.

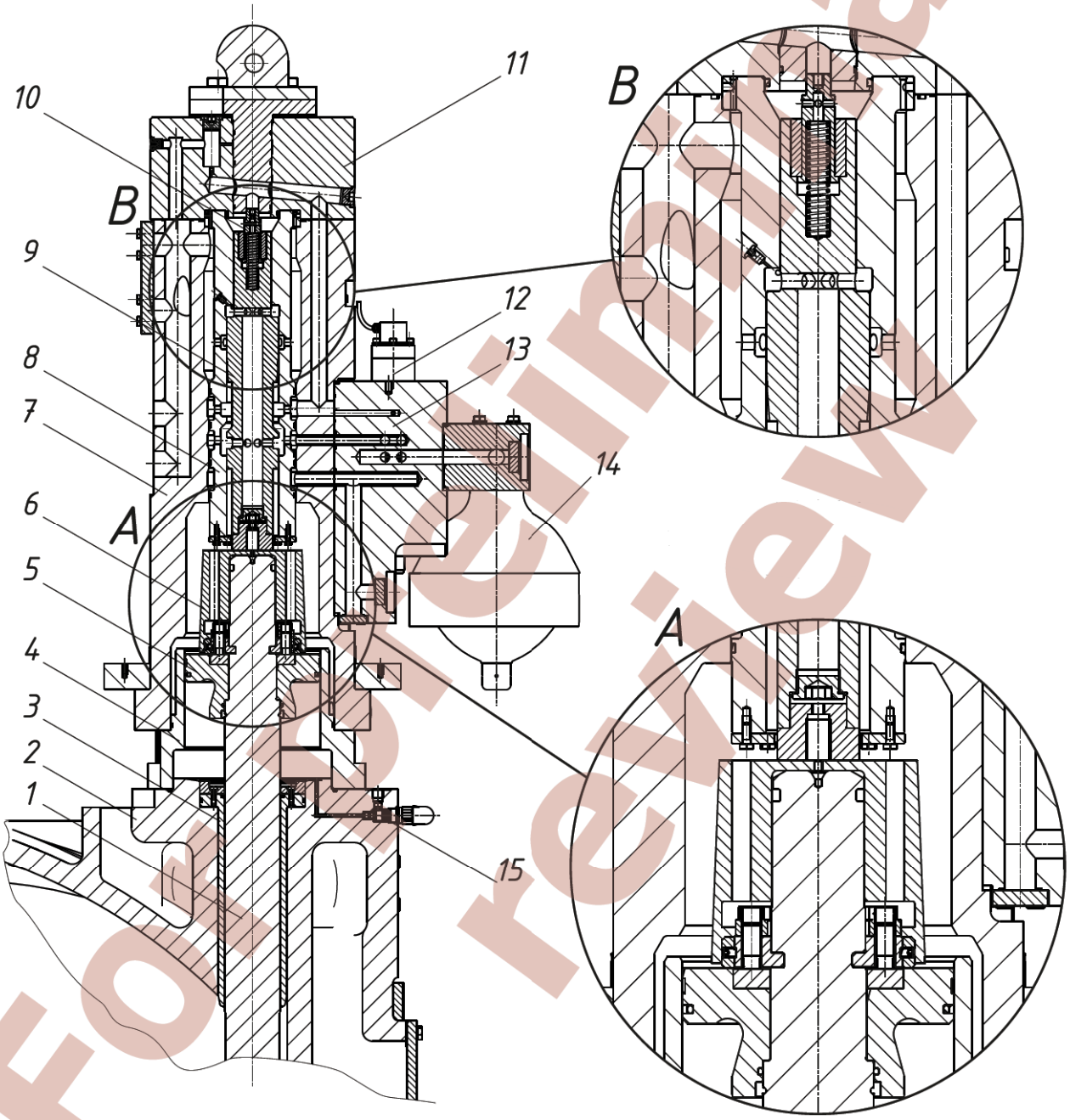


Figure 4.35 – General arrangement of the top-controlled exhaust valve drive TCEV: 1 – valve spindle; 2 – valve assembly housing; 3 – valve guide bushing; 4 – pneumatic spring cylinder; 5 – pneumatic spring piston; 6 – measuring cone; 7 – drive mechanism housing; 8 – control sleeve; 9 – control spool; 10 – valve closing speed limiter; 11 – drive mechanism cover; 12 – solenoid control valve; 13 – fastening plate; 14 – membrane accumulator of control oil pressure; 15 – air spring pressure limiter valve (adapted from [29])

For preliminary review

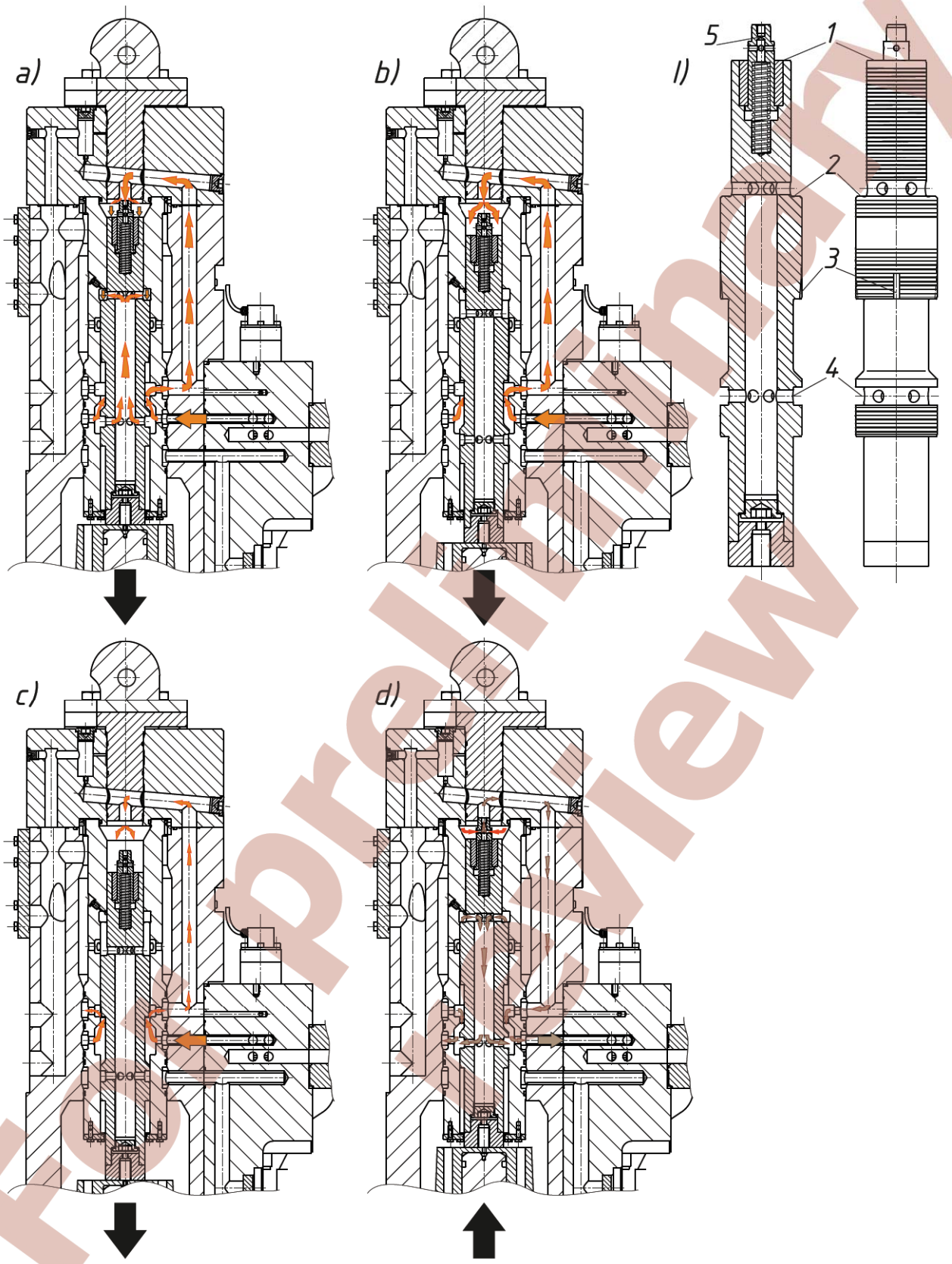


Figure 4.36 – Operating order of the TCEV exhaust valve drive: *a* – start of opening, maximum force, main and additional cavities operate; *b* – main stroke, only the main hydraulic cavity works; *c* – the final phase of valve opening, the flow of oil entering the main cavity is throttled; *d* – valve closing, oil throttling through the closing speed limiter; *e* – spool piston. 1 – working surface of the main hydraulic cavity; 2 – working surface of the additional hydraulic cavity; 3 – throttling grooves; 4 – oil supply to the additional hydraulic cavity; 5 – throttling hole of the valve closing speed limiter (adapted from [29])

For preliminary review

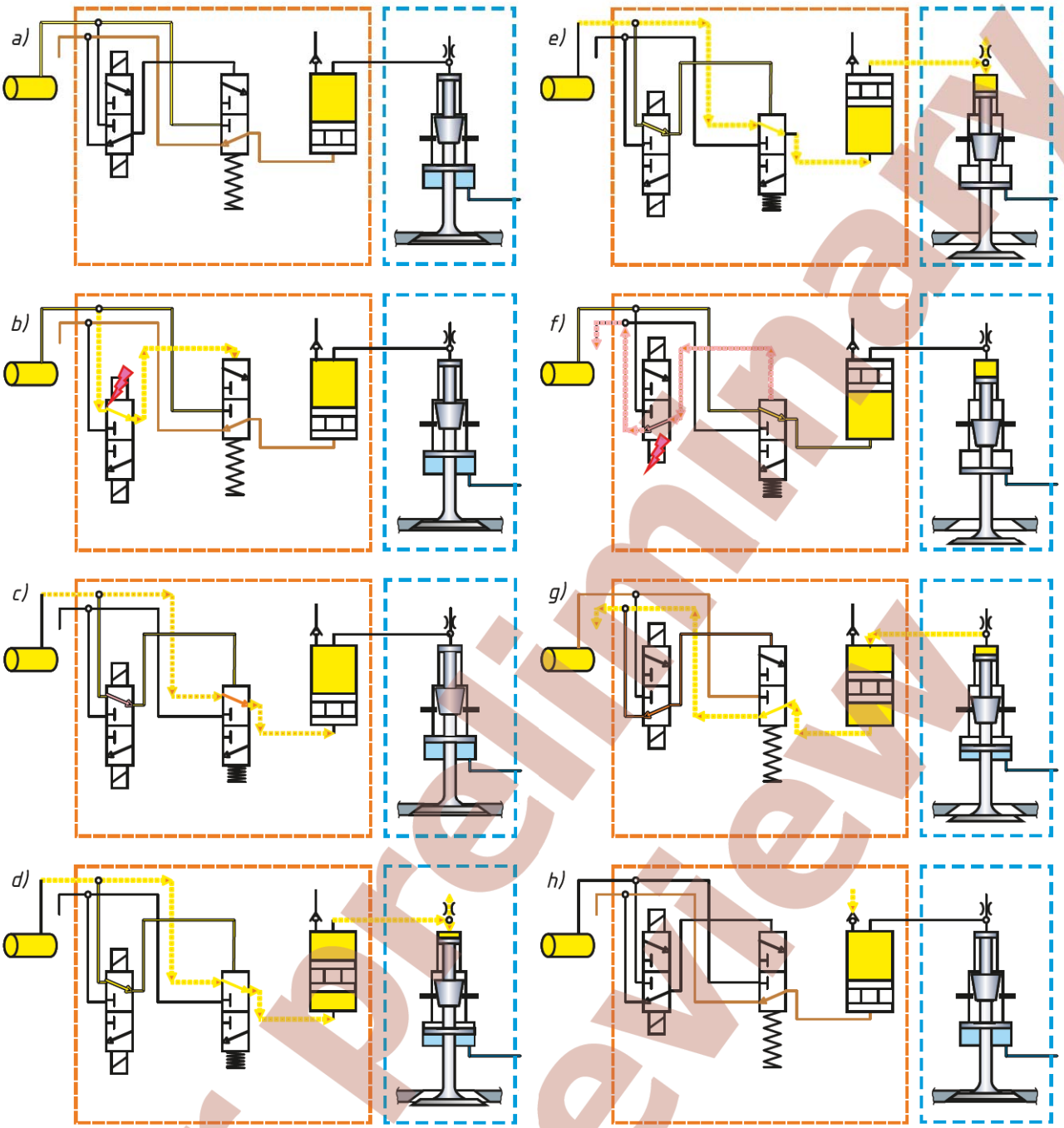


Figure 4.37 – Hydraulic diagram of the exhaust valve drive of a diesel engine of the RT-flex 96C series from Wärtsilä and the order of its operation: *a* – the valve is closed, there is no control action; *b* – control action on the opening of the main valve; *c* – the actuator spool moves and opens the oil supply to the hydraulic cylinder; *d* – the actuator piston moves, the exhaust valve opens; *e* – full opening of the valve; *f* – control action for closing the main valve; *g* – draining oil from the hydraulic cylinder; *h* – replenishment of oil leaks: 1 – control hydraulic line; 2 – main valve; 3 – actuator valve spools; 4 – hydraulic piston of the actuator; 5 – hydraulic piston of the valve drive; 6 – measuring cone; 7 – valve stem position sensors; 8 – pneumatic piston; 9 – exhaust valve; 10 – compressed air supply; 11 – deaeration throttle; 12 – drain line; 13 – make-up valve (adapted from [30])

To overcome the force from the gas pressure in the working cylinder of the engine at the first stage of opening the exhaust valve, a lot of force is required. This is achieved by the fact that the oil entering the executive hydraulic cylinder through a channel in the drive housing is supplied to the main hydraulic cavity located above the upper end surface of the spool piston (Fig. 4.36 *a*), while overcoming the spring force of the valve closing speed limiter 10. Through the holes 4 (Fig. 4.36 *e*) the oil enters the internal channel in the spool piston and then enters the additional hydraulic cavity,

For preliminary review

creating additional force. As the spool piston moves, the edge of the oil chamber cuts off the oil supply through holes 4 (Fig. 4.36 *e*) into the additional cavity (Fig. 4.36 *b*), thereby reducing the force acting on the valve spindle in the area of its main stroke. During the final stage of opening, the valve speed must be reduced to avoid so-called overshoot. That is, the force impact by the time the valve is placed on the stop should be minimal. To ensure this, the spool piston gradually blocks the oil supply to the main hydraulic cavity, throttling the flow using four grooves of variable cross-section 5 (Fig. 4.36 *e*) made on its forming surface (Fig. 4.36 *c*).

At partial loads, when the pressure in the working cylinder is not high enough, the throttling of the oil flow, regulated by the relative position of the piston and the control sleeve, is insufficient to slow down the spindle speed. In this case, throttling of the oil supply to the actuator is used using a PEVA control valve, which has proportional characteristics.

When the valve closes, the hydraulic distribution block connects the oil cylinder of the TCEV actuator to the return line. As a result of this, the valve spindle, under the influence of a pneumatic spring, returns to its original position, moving the spool piston upward and thereby displacing oil from the working cavities.

At the end of the valve closing stroke, the cross-section of the oil channel is blocked by a protrusion on the closing speed limiter 10 (Fig. 4.35), and further displacement of oil occurs through the throttle hole 5 (Fig. 4.36) in its housing. As a result, the closing speed is significantly reduced and the exhaust valve sits on its seat without shock (Fig. 4.36 *d*).

Oil under a pressure of 20 MPa is pumped into the system by axial-piston pumps driven from the engine. For starting and operating at low speeds, the engine is equipped with a pumping station. In a number of designs, to provide feedback between the actuator and the control unit, elements for controlling the movement of the exhaust valve are provided.

In Fig. 4.38 shows the drive mechanism for the exhaust valve of a diesel engine of the RT-flex 96C series with a measuring cone, which is rigidly fixed to the valve stem. To control the position of the rod, two sensors are installed on the valve actuator housing that respond to changes in the gap between the sensor and the measuring cone. The signal from the sensors is sent to the electronic control unit and is used to generate a control signal.

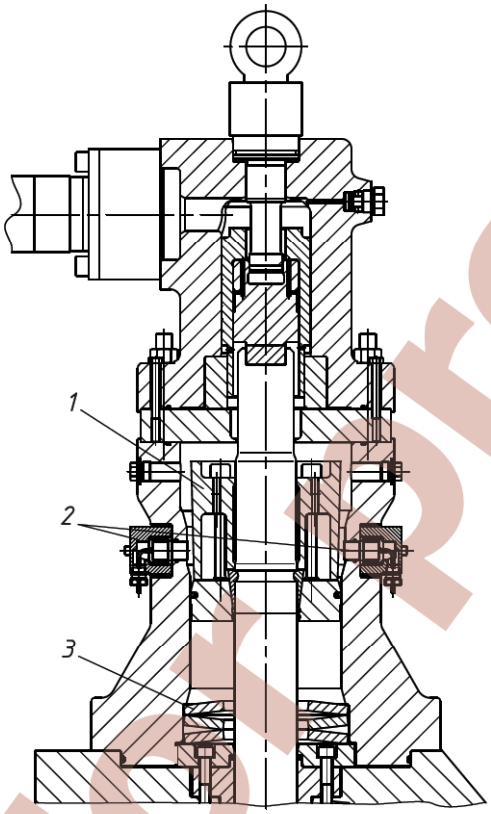


Figure 4.38 – Diesel exhaust valve drive of the RT-flex 96C series from Wärtsilä with a measuring cone: 1 – measuring cone; 2 – valve stem movement sensors; 3 – spring damper of the air piston (adapted from [27])

4.5.9 Changing valve timing when reversing the engine

To ensure efficient operation of a diesel engine when changing the direction of rotation, it is necessary to change the valve timing to mirror opposite ones.

On modern marine low-speed two-stroke diesel engines, the following design schemes are most widespread:

- with cams that have a symmetrical profile and provide equal valve opening and closing angles relative to TDC;
- with asymmetrical phases optimized for forward mode;
- with flexible control of the exhaust valve operating mode through the use of hydraulic actuators

For preliminary review

and electronic control of gas exchange processes.

On four-stroke engines, the valve timing is changed due to:

- installation of a double set of cam washers for forward and reverse with their rearrangement by axial movement of the camshaft;

- turning the cam shaft relative to the crankshaft.

In modern two-stroke LSEs with direct-flow valve purge and having a mechanical gas exchange control system, as a rule, use cams with a symmetrical profile, which ensure symmetrical opening and closing of the exhaust valve relative to TDC, regardless of the direction of engine rotation. This solution makes it possible to significantly simplify the engine design, but does not allow optimizing the gas exchange process when operating in forward gear due to the use of asymmetrical valve timing. If, however, the engine is set to open and close the exhaust valve earlier in forward mode, then when reversing, the conditions for purging the working cylinder worsen and the engine's operating efficiency decreases. In some cases, for such engines, manufacturers limit the load and operating time in reverse.

The simplest task of changing valve timing during reversal is implemented in engines with hydraulic drive of actuators and electronic control of operating processes. In such engines, changing the valve timing is carried out by changing the algorithm for supplying control electrical signals by a microprocessor controller to the gas exchange control system. In this way, optimal opening and closing angles of the exhaust valve can be ensured in forward and reverse modes.

Four-stroke engines typically use two sets of cams, one for forward operation and the other for reverse. The shims are rearranged by axial movement of the camshaft (Fig. 4.39). To do this, a special servo drive is installed on the engine, consisting of a hydraulic cylinder and a piston located in it. The supply of oil to one of the cylinder cavities ensures the movement of the piston in one direction or another with the installation of the corresponding set of cam washers under the pusher rollers.

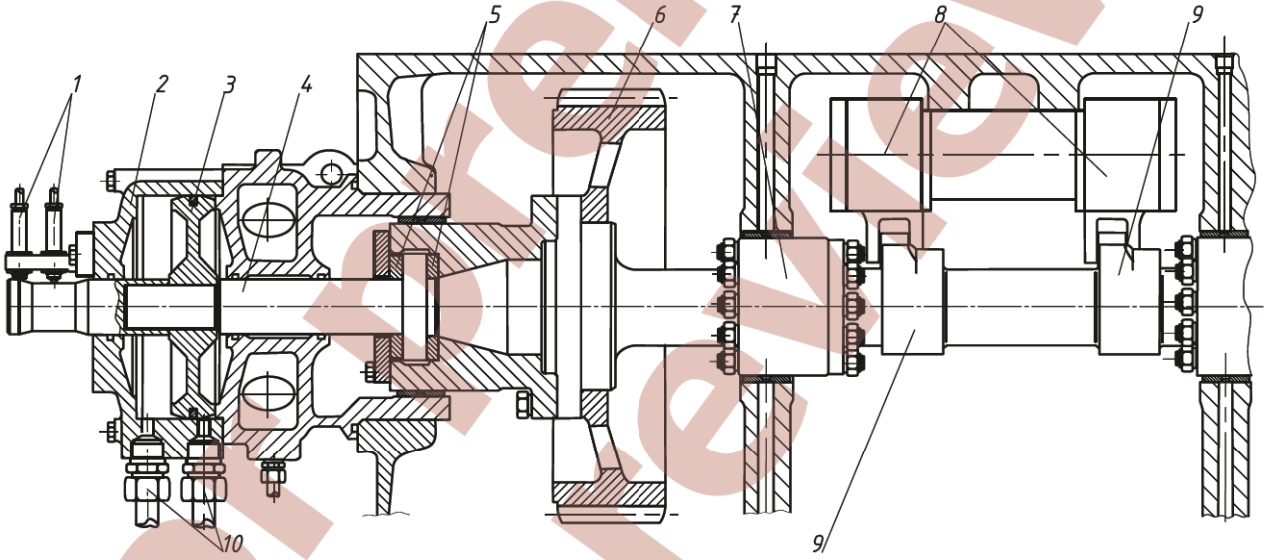


Figure 4.39 – Mechanism for changing valve timing when reversing a medium-speed engine: 1 – camshaft position sensors; 2 – hydraulic cylinder; 3 – piston for axial movement of the camshaft; 4 – idler shaft; 5 – thrust bearings; 6 – camshaft drive gear; 7 – camshaft; 8 – axis of lever pushers; 9 – double cams with profiles for forward and reverse; 10 – control oil supply (adapted from [31])

For shock-free transfer of rollers from one set of washers to another, a special inclined transition profile is made or a mechanism is provided for lifting and fixing the pushers while the camshaft moves.

When the camshaft rotates relative to the crankshaft, the drive mechanism is equipped with a servo drive with angular movement. After submitting a control signal, the servo drive rotates the camshaft relative to the drive gear at an angle of 90°, thereby ensuring a change in valve timing to the opposite.

For preliminary review

4.5.10 Changing valve timing to optimize engine operating parameters

It should be understood that the valve timing adopted for each specific engine and specified by the profiles of the cam washers, as well as their wedge angles, are not optimal for the entire range of engine operating conditions. Typically, valve timing optimization is carried out for the most frequently used engine operating mode, most often the rated power mode. Deviation from this mode leads to a deterioration in the operating process conditions for a number of reasons, including due to a deterioration in the filling of the cylinders due to deviation of the valve timing from optimal ones. These deviations lead to an increase in the content of harmful components and, first of all, soot particles in the exhaust gases. It should be noted that engine operation at low loads is usually associated with the navigation of ships in coastal waters, when calling at ports, passing through canals, that is, areas where particularly high environmental requirements are imposed. In this regard, many manufacturers are considering the possibility of changing valve timing depending on the load and speed conditions of the engine as an important direction towards further improvement of marine engines. Moreover, the ability to change valve timing is included not only in the design of new engines, but also devices are being developed that make it possible to modernize engines already in operation without significant costs.

Four-stroke engines operating on the Miller cycle (see paragraph 2.4.2) have good operating performance at medium and high loads. This is ensured by a sufficiently high boost pressure through the use of highly efficient turbochargers. However, when switching to low loads, the boost pressure decreases sharply and, accordingly, the filling of the cylinders with fresh air deteriorates, which leads to a deterioration in fuel combustion conditions. To ensure a more complete charge of the cylinder at low loads, it is rational to increase the dynamic charging stage (see paragraph 2.4.2) by closing the intake valve later, that is, switching from the Miller cycle to the classical cycle. It is this principle that is implemented in most modern devices that control valve timing. In this regard, the possibility of changing the opening and closing angles of the intake valve is most often considered, but new designs provide for the ability to control the exhaust valve, also changing the angle of valve overlap to improve the conditions of purging the combustion chamber or to reduce losses of charge air.

Conventionally, all mechanisms that allow changing valve timing can be divided into three groups: mechanical, hydromechanical and electrohydraulic.

Mechanical devices include devices that allow you to rotate the camshaft relative to the crankshaft at a certain angle, or change the position of the pusher relative to the cam washers, thereby changing the angles at which the valves begin to open and close.

The device, developed by MaK and called Flex Cam Technology (FCT), is shown in Fig. 4.40 *a*.

The principle of its operation is based on changing the position of the lever pusher roller relative to the camshaft cam washer. To do this, the lever axis is mounted eccentrically on a shaft, which has the ability to rotate through an angle close to 180°. As a result, the pusher makes a translational movement, changing the angles at which the valve begins to open and close.

In various versions of this device installed on the engine, the valve timing is controlled for the intake valves (Fig. 4.40 *b*) or the intake and exhaust valves. In combination with control of the fuel injection angle, this system makes it possible to ensure compliance with current soot emission standards over the entire range of engine loads (Fig. 4.40 *c*).

The eccentric shafts of the roller pushers are driven by a pneumatic servomotor through a gear system. Manual switching of the engine from one mode to another is also provided. Currently, only stepwise control of valve timing is used, but in the future it is also possible to smoothly change the opening and closing angles of the valves depending on the operating mode.

As in the first case, this device does not allow you to change the duration of valve opening. This disadvantage is absent in the hydromechanical devices shown in Fig. 4.40 *d*, *e* which were called Variable Inlet Valve Closing (VIC)

The device, the diagram of which is shown in Fig. 4.40 *d*, designed for the modernization of Sulzer ZA40S engines in service (Fig. 4.41). It consists of two hydraulic cylinders separated by a

For preliminary review

valve, in the plate of which a throttle channel is provided. The valve has an external pneumatic drive.

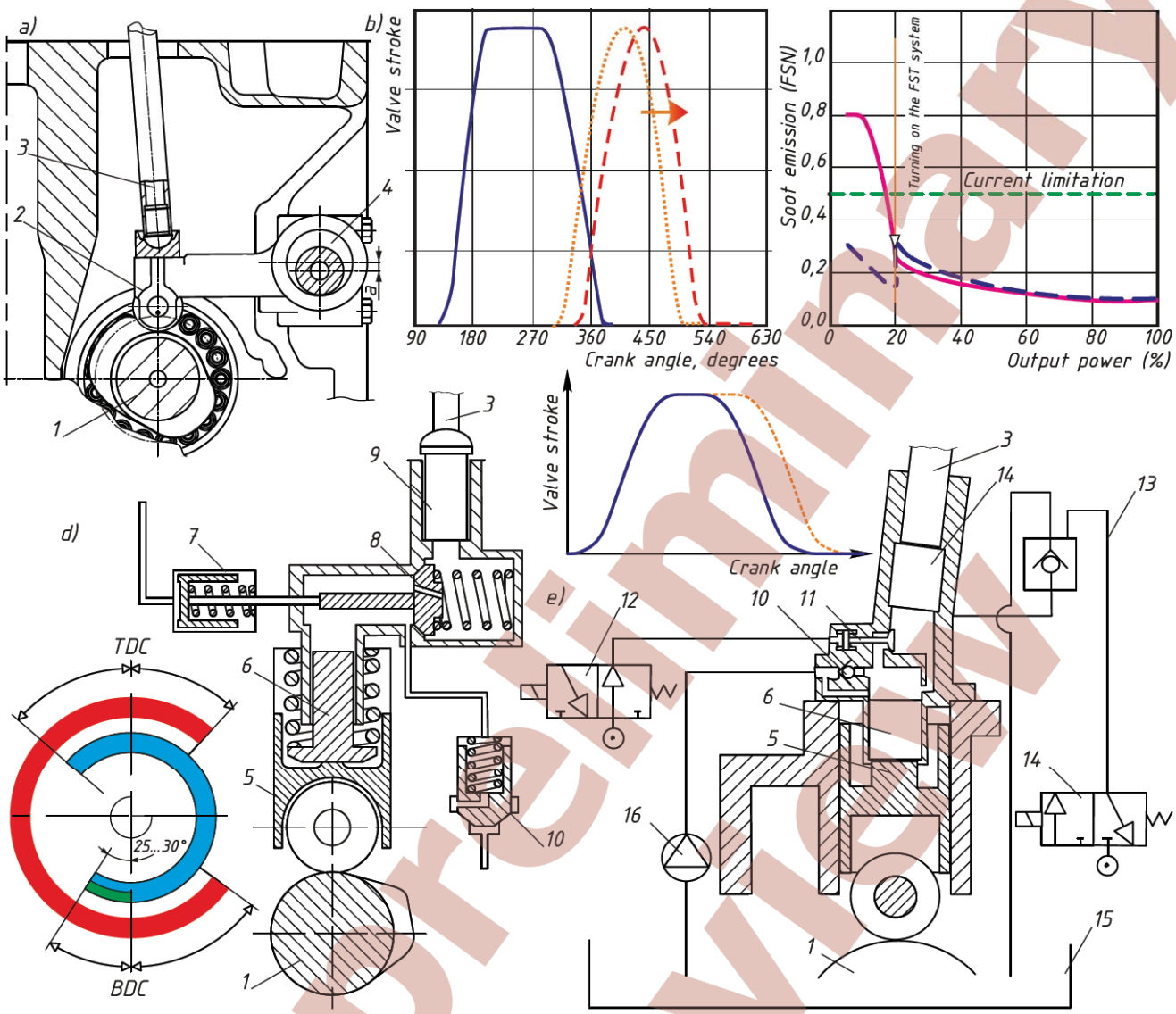


Figure 4.40 – Mechanisms for changing valve timing: *a* – with movement of the lever pusher relative to the cam washer (MaK engines); *b* – change in the angles at which the intake valve begins to open and close in MaK engines; *c* – reduction in exhaust gas smoke with increasing dynamic purging phase in MaK engines; *d* – hydromechanical drive with fluid throttling of Sulzer ZA40S engines; e – hydromechanical drive with control valve for Wärtsilä 46F series engines: 1 – camshaft; 2 – roller pusher; 3 – rod; 4 – eccentric of the lever axis; 5 – pusher; 6 – working piston; 7 – pneumatic drive of the separation valve; 8 – separating valve; 9 – executive piston; 10 – make-up valve; 11 – control valve; 12 – control valve spool; 13 – deaeration valve; 14 – drain valve; 15 – oil storage tank; 16 – charging pump (adapted from [14, 31, 32, 33])

One cylinder contains a piston that receives forces from the pusher and transmits them to the liquid. The second cylinder contains an actuator piston, which receives forces from the fluid and transfers them to the valve drive rod. At medium and high loads, the valve separating the cavities of both pistons is kept open and does not interfere with the oil circulating freely between them. When switching to low load mode, the air holding the valve is released, and the valve sits on its seat, separating the piston cavities. In this case, when the pusher moves upward, the valve opens under pressure and almost freely allows oil to pass into the cavity of the actuating piston. During the reverse stroke, the valve closes and the oil from the cavity of the slave cylinder returns to the working cylinder through the throttling channel in the valve disc. As a result, the valve closure occurs with a

For preliminary review

delay, which leads to an increase in the time it takes to fill the working cylinder of the engine with fresh charge.

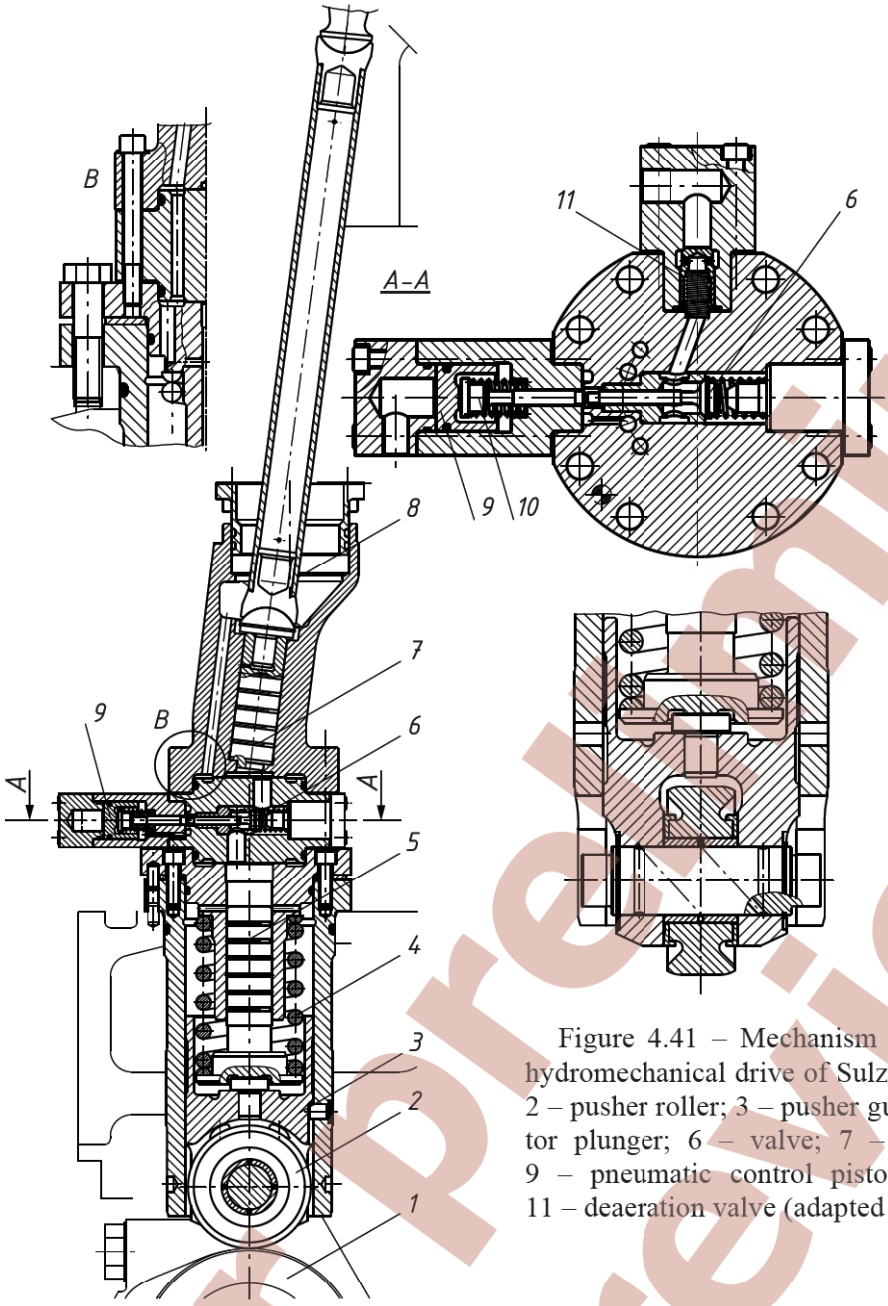


Figure 4.41 – Mechanism for changing valve timing with hydromechanical drive of Sulzer ZA40S engines: 1 – camshaft; 2 – pusher roller; 3 – pusher guide; 4 – return spring; 5 – actuator plunger; 6 – valve; 7 – pusher rod; 8 – drive piston; 9 – pneumatic control piston; 10 – control valve pusher; 11 – deaeration valve (adapted from [34])

Wärtsilä has developed a device for the series engines 46F, the diagram of which is shown in Fig. 4.40 *f*. Structurally, this device is similar to the previous one, with the only difference being that the hydraulic cavities are connected by two channels. The flow of fluid through one channel, starting at the base of the cylinder, is controlled by a pusher piston, and the other flow is controlled by a valve having an independent hydraulic drive connected to the engine management system. If the control valve is open, oil circulates freely between the cavities and the control piston exactly follows the movement of the pusher. When the control valve is closed, the lowering of the actuating piston begins only after the pusher piston opens the second channel connecting the cavity. As a result, valve closure is delayed, which can be up to 30° according to the crankshaft angle.

In two-stroke engines with mechanical control, variable valve timing systems are also used, which serve to increase the efficiency of the operating process at low load conditions. When the diesel engine speed decreases, due to a decrease in boost pressure, the weight filling of the working cylinder decreases. To maintain high values of the parameters of the working fluid at the end of compression, which is necessary for successful combustion of fuel, in some types of LSE, the actual

For preliminary review

compression ratio is increased when the load is reduced. This is accomplished by increasing the piston stroke by closing the exhaust valve earlier, almost simultaneously with closing the purge ports. In Fig. 4.42 shows a diagram of the hydromechanical control system for the exhaust valve LSE type RTA84T from Wärtsilä, called Variable Exhaust valve Closing (VEC).

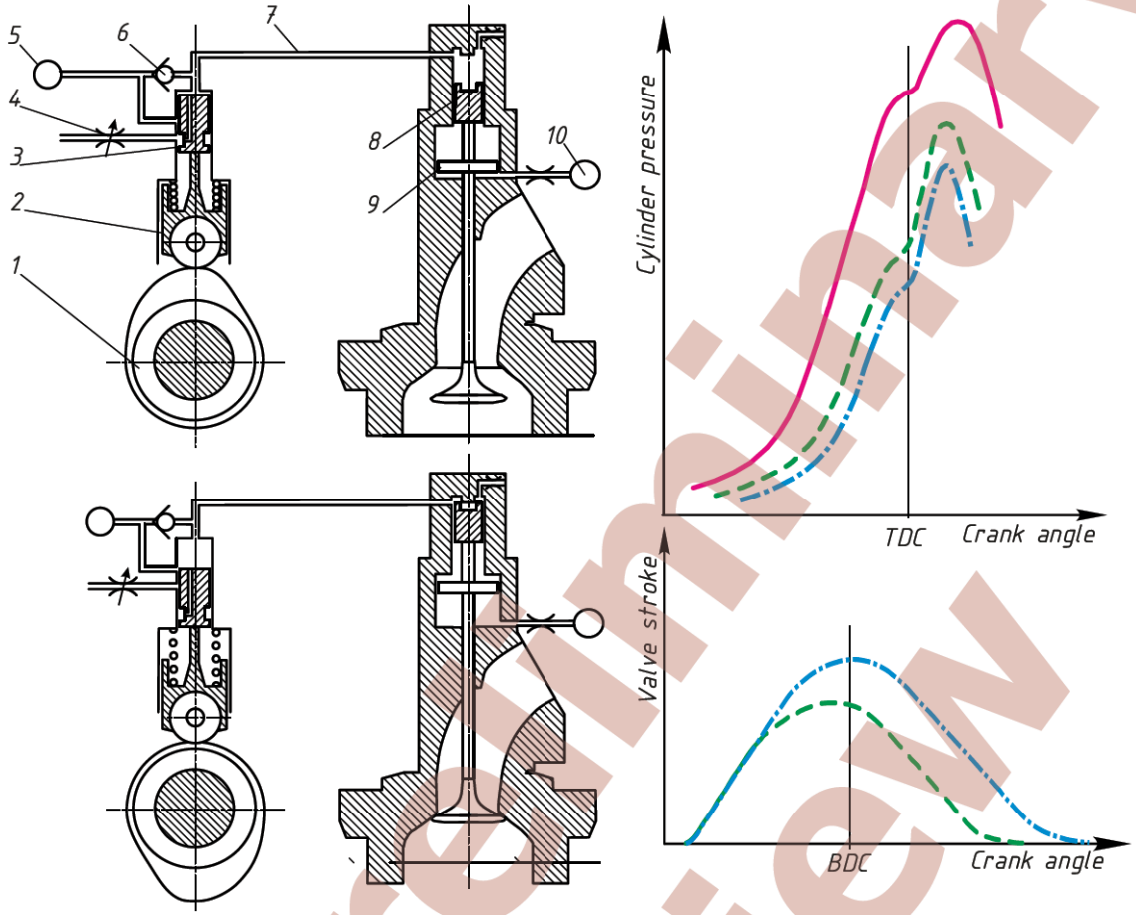


Figure 4.42 – Mechanism for changing the exhaust valve closing angle (VEC) of the RTA84T engine: 1 – camshaft cam; 2 – roller pusher; 3 – actuator piston; 4 – VEC control valve; 5 – hydraulic line; 6 – check valve; 7 – high pressure hydraulic line; 8 – hydraulic piston of the actuator; 9 – air piston; 10 – air line. – · – · – partial load without VEC; – – – – partial load with VEC; ——— – full load (adapted from [35])

To achieve earlier closing of the valve, an annular groove is made on the actuator piston, which communicates with the above-piston space through a channel inside the piston. At the bottom of the actuator hydraulic cylinder there is an outlet channel, closed by a control throttling valve.

If the valve is closed, the oil cannot leave the space above the piston and enters in full into the actuator, the piston of which exactly repeats the movement of the pusher. When the load decreases, the engine control system opens the control valve and as soon as the groove on the actuator piston coincides with the outlet channel, oil begins to flow out of the space above the piston through the throttle channel in the control valve. The exhaust valve begins to close even before the actuator piston reaches the upper position. In Fig. 4.42 shows the influence of the position of the control valve on the law of valve movement and the nature of the operating process in the engine.

The principle of valve timing control is most widely implemented in engines with electronic valve control. They can use flexible control schemes, selecting the most optimal valve timing in accordance with the engine operating mode (Fig. 4.43 *a*). Thus, for engines of the ME series from MAN, when the load is reduced, not only earlier closing of the exhaust valve is provided, but also its earlier opening (Fig. 4.43 *b*).

This allows gases to be released from the cylinder under higher pressure, which in turn improves the operating conditions of the turbocharger and makes it possible to increase the weight charge of the cylinder at medium and low load conditions.

For preliminary review

4.6 Elements of the air-gas system for receiving and purifying air

In most cases, the engine air intake is carried out from the engine room (Fig. 4.44 *a*). This allows you to obtain a more stable air temperature at the inlet, as well as improve the ventilation conditions of the ship's engine and boiler room.

The disadvantages of this method of sampling are:

- a fairly high temperature of the air entering the engine, which reduces its density and, consequently, the weight filling of the cylinders;
- the presence in the air of the engine room of aerosol droplets of various technical liquids, primarily fuel and oil, which leads to rapid clogging of filters and the flow path of turbochargers, intake channels and other gas distribution parts.

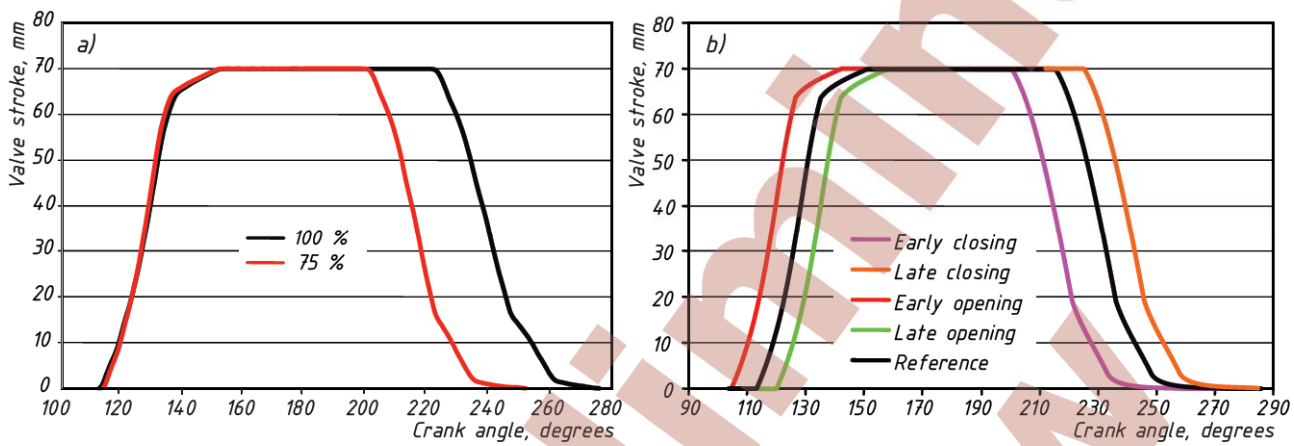


Figure 4.43 – Changes in the opening law of the exhaust valve of the low-speed two-stroke engine S50ME-C depending on its operating mode (adapted from [36])

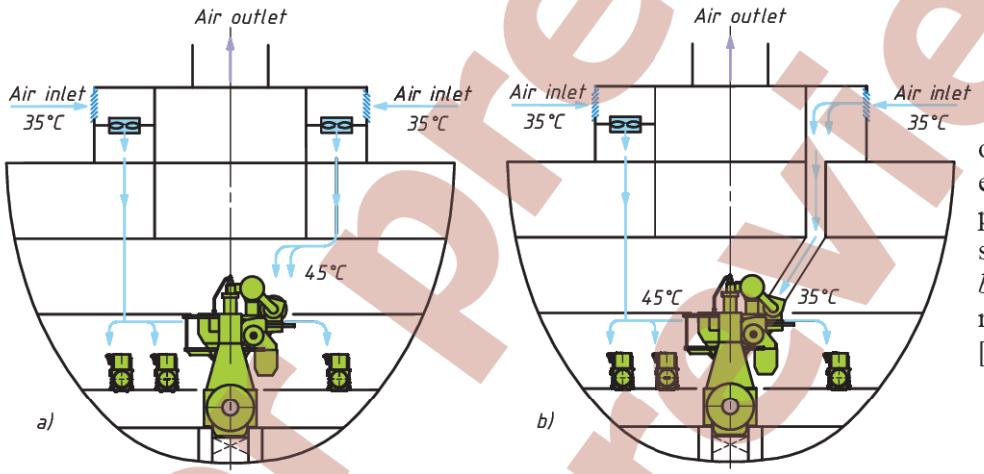


Figure 4.44 – Methods of air intake for the main engines of the ship's power plant: *a* – from the ship's engine room; *b* – directly from the environment (adapted from [37])

Some modern types of ship power plants use a circuit with outside air intake (Fig. 4.44 *b*). For this purpose, special air-conditioning is carried out in the machine-boiler room. The outside air temperature is usually significantly lower than the temperature in the engine room, which improves the air supply conditions and the performance of the entire energy installation.

The disadvantages of this method of air intake include:

- more complex design;
- the presence of sea salt particles in the air, which enhance high-temperature sodium-vanadium corrosion;
- large fluctuations in outside temperature depending on both the navigation area and the time of day, which requires the installation of additional control devices.

As already noted, the main air pollutants for marine engines are aerosol particles of fuel and oil, the content of which usually ranges from 0.5...5 mg/m^3, and in some cases can reach 20...30 mg/m^3.

For preliminary review

The share of other types of particles (dust, soot) in the total volume of air is disproportionately small. Air contamination primarily depends on the location of its intake, as well as on the volume of the engine room, its saturation with mechanisms, the frequency of air circulation, and the specific power of the mechanisms per 1 m^3 volume of the engine room.

Air purification is carried out using air purifiers, which can be divided into two groups with dynamic action and filtering action.

Dynamic purifiers include devices that use differences in the mass of these particles and air molecules to remove polluting particles. Their operation is based on the principle of the influence of dynamic forces on particles during a sharp change in the direction of the flow or when it twists. Air molecules, being lighter, manage to change the direction of their movement, and polluting particles, being heavier, continue to move by inertia and, hitting the surface of the purifier, lose speed, sticking to it. In a number of designs called *cyclones*, the air is given a rotational motion, in which a centrifugal force acts on the particles, pressing them against the wall of the purifier. Dynamic ones include baffle cleaners with an oil bath, direct-flow and counter-flow cyclones, and mesh cleaners. The advantage of such devices is their design simplicity and reusability, as well as low input resistance. In addition, such purifiers, having a relatively large volume, smooth out air pulsations, reducing suction noise and improving the operating conditions of the supercharger compressor. The disadvantage of dynamic cleaners is their relatively low cleaning efficiency; they are capable of retaining no more than 95% of polluting particles. The remaining particles entering the intake tract lead to its rapid clogging. Therefore, dynamic filter elements are currently used only for air pre-cleaning.

On modern engines, dry filter elements are widely used, the basis of which is the principle of trapping polluting particles when air passes through a microporous partition, which is used as felt, special cardboard or other non-woven materials. The pore sizes for a filter element made of cardboard are in the range of 2.5...3.5 μm, for non-woven material 12...15 μm, which allows you to retain 98.5...99.9% of polluting particles. As a result, the wear of CPG parts is reduced by 20...30%, and clogging of the flow part of the intake tract occurs much more slowly. The design of a dry cylindrical filter is shown in Fig. 4.45.

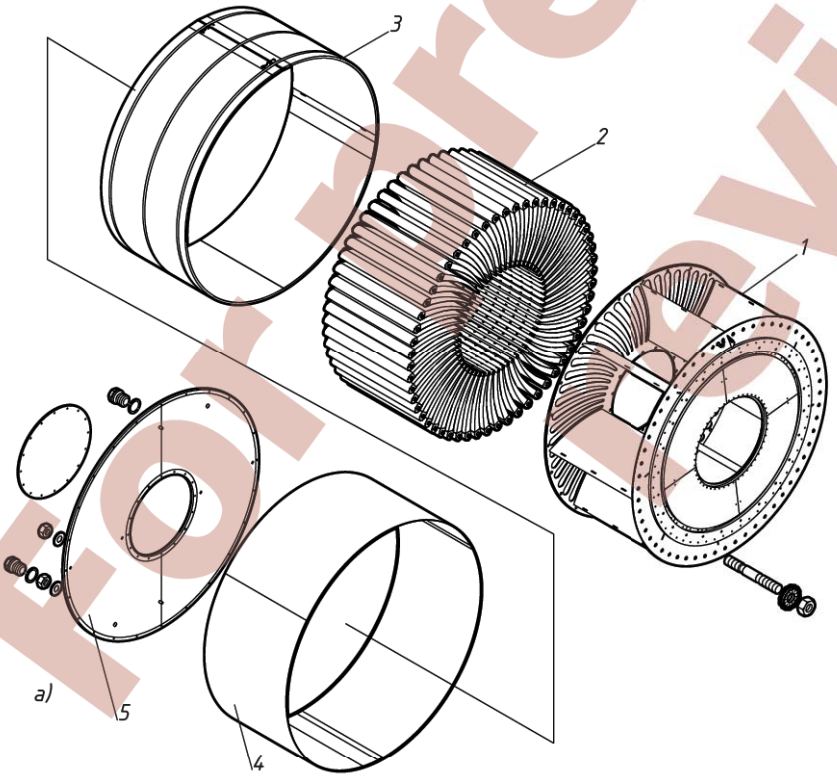


Figure 4.45 – Cylindrical dry filter element (*a*) and filter paper structure (*b*). 1 – frame; 2 – cardboard corrugation; 3 – metal mesh; 4 – felt mat; 5 – side cover (adapted from [38])

A corrugation made of special cardboard is placed on the filter frame, which allows you to increase the filter surface area many times. Along the edges the corrugation is glued to the side walls of the frame. To prevent the corrugation from collapsing, several stiffening ribs are made on the

For preliminary review

frame, and to prevent the individual surfaces of the corrugation from sticking together, embossments in the form of small protrusions are made on the cardboard. To increase the service life of the paper element, it is covered with a mesh on which a mat of special felt is placed, which acts as the first stage of filtration. To prevent mechanical damage, a protective grille is placed on top of the filter. Fittings are installed in the filter housing for connecting a differential pressure gauge, which allows you to control the degree of filter clogging.

The disadvantage of dry filters is their increased suction resistance, which increases as the filter becomes clogged, as well as the impossibility of repeated use. During operation, the filter elements are washed in a special solution, and after several washes they are replaced with new ones.

4.7 Supercharging units

4.7.1 History of the creation and development of marine diesel supercharging units

Over the past decades, the air-gas systems of marine diesel engines have undergone significant changes, primarily associated with the widespread introduction of gas turbine supercharging (turbocharging) units, which provide a significant improvement in the weight, size, efficiency and environmental performance of the combined engine. Today, almost all marine diesel engines are manufactured with gas turbine supercharging and are equipped with devices for regulating the temperature and humidity of the incoming air.

Pressurization units today have become an integral part of the air-gas system of modern marine diesel engines.

Attempts to increase power and other efficient indicators of piston engines using pre-compression of air in front of the working cylinder were made by many specialists at the dawn of this type of engine. So Gottlieb Daimler in 1885 received German patent DRP No. 34926 in which he proposed an engine with pre-compression of air in the sub-piston cavity (Fig. 4.9). Rudolf Diesel pointed out the advantages of supplying pre-compressed and cooled air to the cylinders in 1896, however, his experiments using the sub-piston cavity as a supercharging compressor did not give positive results.

It became possible to fully realize the benefits of supercharging only after on November 16, 1905, Swiss engineer Alfred Buechi received German patent DR P No. 204630 for a machine installation in which a multi-stage axial compressor and a four-stroke piston engine are connected to each other by a common shaft and a multi-stage turbine operating on engine exhaust gases. This patent is considered to be the beginning of the history of turbochargers powered by exhaust gases (Fig. 4.46).

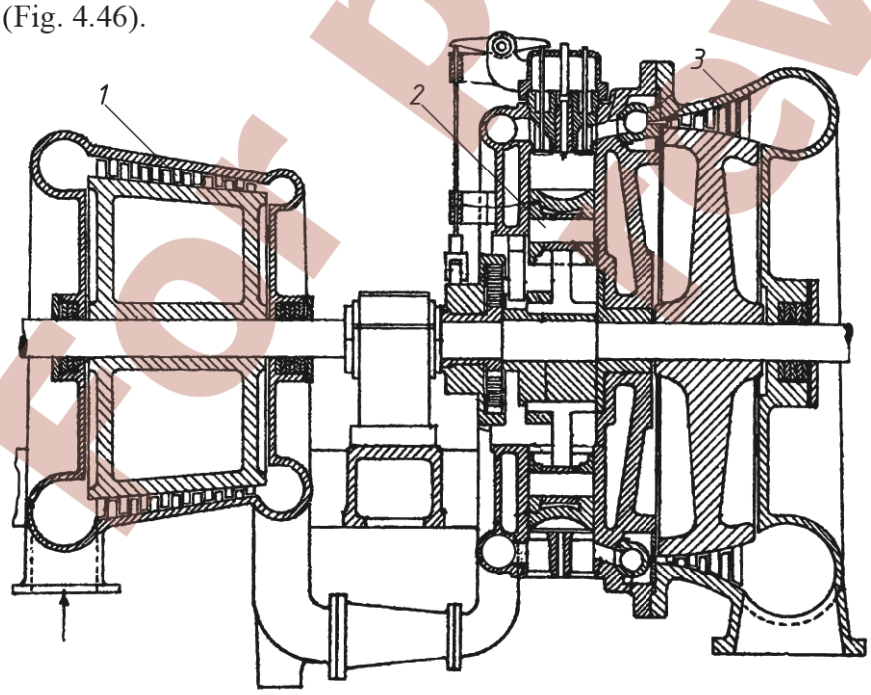


Figure 4.46 – Combination turbopiston engine designed by A. Büchi (patent DR R No. 204630): 1 – air compressor; 2 – piston engine; 3 – gas turbine (adapted from [39])

For preliminary review

Attempts to practically implement the engine have shown a number of difficulties associated with significant differences in the effective rotation speeds of the piston part of the engine, gas turbine and compressor. It was possible to eliminate these contradictions by separating the turbine and compressor into a separate unit that does not have a mechanical connection with the piston part of the engine. Alfred Büchi patented such a unit, which later received the name turbocharger, in 1915 (Fig. 4.47).

For his invention, he receives German patent DRP No. 454107 for a «Composite engine with a high degree of supercharging», consisting of a diesel engine and a radial-axial compressor and an axial turbine separately mounted on a common shaft. It was assumed that the pressure at the outlet of the compressor should be higher than the pressure in front of the turbine, and the turbine should operate in pulse mode.

In 1923, two passenger ships «Preussen» and «Hansestadt Danzig» were launched in Germany, each equipped with two four-stroke ten-cylinder Vulkan-MAN engines equipped with Brown Boveri turbochargers with constant pressure in front of the turbine. The turbocharger units were mounted separately from the engines, which increased pressure losses in the pipelines, but even in this form it was possible to increase the power of diesel engines from 1288 to 1840 kW, that is, by more than 40%. From this moment on, the use of gas turbine supercharging began to be widely introduced on four-stroke diesel engines throughout the world.

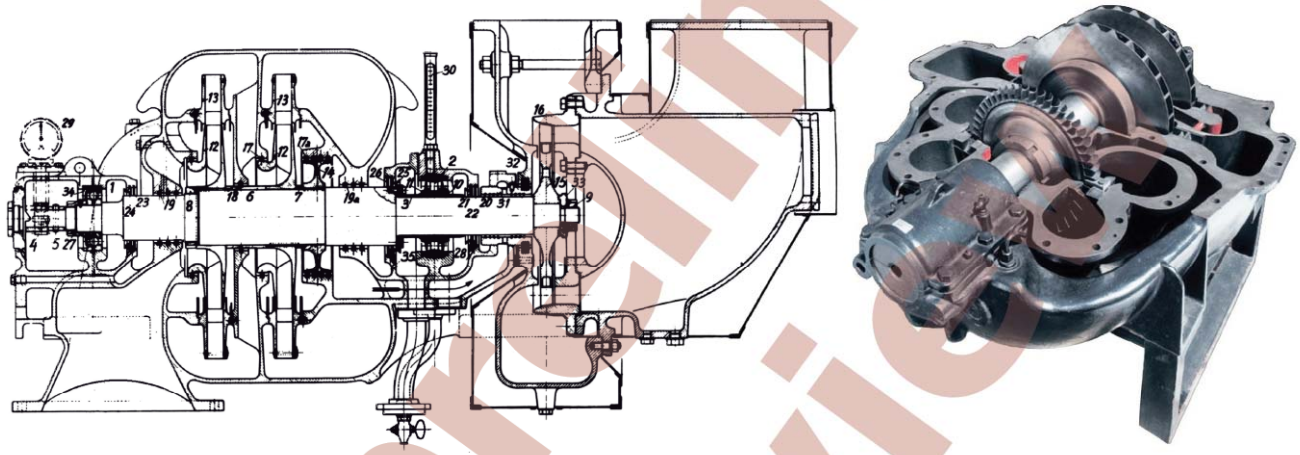


Figure 4.47 – Turbochargers developed by A. Buchi (adapted from [40, 41])

Two-stroke engines have long stood aloof from the use of gas turbine supercharging, although work on the use of turbochargers in such engines has been carried out since 1927. The peculiarities of organizing gas exchange in two-stroke marine diesel engines did not allow for a long time to solve the problem of using gas turbine supercharging. In such engines, mounted piston pumps were used to purge the working cylinder, and subsequently piston pumps were used as additional inflating units.

In the design of a crosshead engine, in which the cylinder is separated from the crankcase by a diaphragm, the possibility of using the sub-piston space as sub-piston pumps was used. Automatic air valves were used to control the flow of air charge between the sub-piston cavity, the working cylinder and the environment. Subsequently, with the advent of gas turbine supercharging on low-speed diesel engines, piston pumps were used as a second compression stage, which was connected in parallel or in series with the turbocharger, pumping air into the common receiver of the second stage or into a special buffer space separate for each cylinder.

The practical result of using gas turbine charging in two-stroke diesel engines was achieved only in 1952. This year, the «Dorthe Maersk» tanker with a displacement of 18,000 tons, built at the AP Moller shipyard in Denmark, entered service. It was the first ship to have a Burmeister & Wain (B&W) two-stroke six-cylinder diesel engine with gas turbine supercharging. Two VTR630 series turbochargers located on the engine increased its power from 4070 to 5800 kW. The tanker Dorthe

For preliminary review

Maersk marked the first milestone in the use of turbocharging for two-stroke marine engines.

Currently, supercharging, in combination with cooling the air entering the working cylinders, is one of the main ways to increase the power of marine diesel engines. It allows you to increase the mass of the air charge entering the working cylinders, which in turn makes it possible to increase the cyclic fuel supply. All other things being equal, the use of supercharging and charge cooling leads to an increase in the power of individual cylinders and the entire diesel engine as a whole.

4.7.2 Methods of supercharging marine engines

All methods of supercharging marine engines known to date can be conditionally divided according to the method of driving the compressors:

- mechanically driven supercharging systems, in which the power to drive the compressor is transmitted from the diesel crankshaft (Fig. 4.48 *a*) through an overdrive called *a multiplier* or from an external source, for example from an electric motor (Fig. 4.48 *b*);

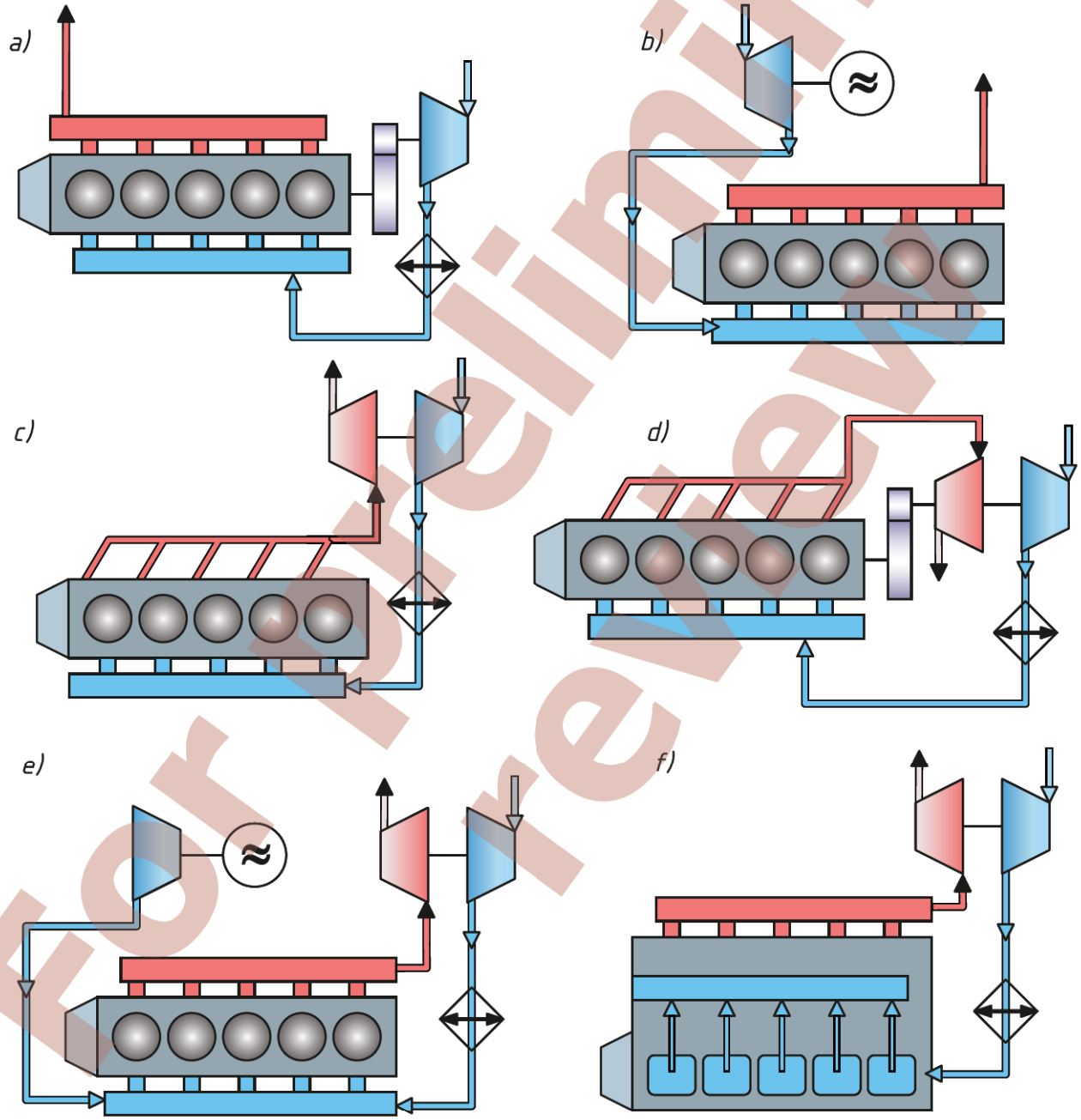


Figure 4.48 – Methods of supercharging marine diesel engines: *a* – mechanical supercharging; *b* – boost from an electric compressor; *c* – gas turbine supercharging without mechanical connection; *d* – gas turbine supercharging with mechanical connection; *e* – combined boost; *f* – boost using sub-piston cavities. ◀ – exhaust gases; ◁ – air

For preliminary review

- supercharging systems with a gas turbine compressor drive, in which the gas turbine uses the energy of the exhaust gases and drives a compressor located with it on the same shaft, forming a single unit called a *turbocharger*. In this case, the turbocharger shaft does not have a mechanical connection with the engine shaft (so-called free turbocharging) (Fig. 4.48 *c*), or is connected to the engine through a mechanical transmission (Fig. 4.48 *d*);

- combined pressurization using both drive and gas turbine compressors (Fig. 4.48 *e*).

Early marine engine developments used mechanical charging systems using both positive displacement and dynamic compressors. The presence of a mechanical connection between the compressor and the crankshaft makes it quite easy to match the characteristics of the compressor with the flow characteristics of the piston part of the engine, especially in transient operating conditions. In a number of designs, not only a rigid but also a flexible mechanical connection was used for this purpose, allowing the compressor speed to be changed, regardless of the crankshaft speed. The main disadvantage inherent in this method of supercharging is a decrease in the mechanical efficiency of a diesel engine associated with the expenditure of 7...10% of the energy generated by the engine to drive the compressor, and as a result, increased fuel consumption. In addition, a significant increase in air pressure required unjustified complication of the design of both the drive and the compressor itself.

Due to these shortcomings, all leading manufacturers of four-stroke marine diesel engines have switched to using free turbocharging as the main way to increase engine power and efficiency.

In two-stroke crosshead engines, combined charging schemes were used for a long time, when sub-piston cavities were used as one of the stages, operating as piston pumps (Fig. 4.44 *f*). Subsequently, with the improvement of gas turbine charging systems, the use of sub-piston cavities was abandoned.

Currently, two-stroke LSEs use a combined charging scheme, which includes a turbocharger that supplies the engine with air in medium and full power modes, as well as electrically driven blowers that provide operation in start-up modes and low loads (Fig. 4.44 *e*). In addition, blowers ensure diesel operation in emergency modes, for example, in the event of a turbocharger failure.

Thus, gas turbine supercharging today is an integral part of modern marine diesel engines. Many years of design and operation experience have shown that increasing the mass of the air charge is the most efficient use of exhaust gas energy. Other methods of supercharging are currently used extremely rarely or in combination with the same gas turbine supercharging.

4.7.3 Gas turbine charging

Gas turbine supercharging or turbocharging is a method of supercharging based on the use of energy from the exhaust gases of the piston part of the engine. The main unit is a turbocharger that has no mechanical connection with the engine shaft. The use of gas turbine charging makes it possible to increase engine power by 1.5...4 times.

The widespread introduction of gas turbine supercharging in marine diesel engines is explained by a number of advantages that these engines have compared to naturally aspirated ones:

- at a constant rotation speed, the use of boost makes it possible to increase engine power almost in proportion to the increase in boost pressure;

- the use of supercharging can significantly improve the weight and dimensions of both the engine itself and the ship's power plant as a whole;

- increasing the air pressure at the engine inlet using turbocharging leads to an increase in the share of useful work in the overall energy balance, resulting in an increase in engine efficiency and a decrease in specific fuel consumption (Fig. 4.49);

- the presence of a gas turbine significantly reduces the exhaust noise;

- inflatable engines are less sensitive to changes in external conditions, primarily to changes in temperature, pressure and humidity of the surrounding air;

- with the same method of organizing the operating process, supercharged engines have better environmental performance.

The disadvantages of turbocharging are the increase in mechanical stress of the engine and the

For Preliminary review

difficulty of matching the characteristics of the turbine, compressor and piston part. At the same time, the thermal stress of inflatable engines increases slightly compared to naturally aspirated engines.

Currently, the possibilities of supercharging are limited by the maximum permissible level of thermal and mechanical stress of a diesel engine. In modern four-stroke MSE and HSE, the average effective pressure at rated power reaches 1.7...2.8 MPa. Compared to naturally aspirated diesel engines, for which p_e lies within 0.5...0.6 MPa, the increase in power is 3.4...4.7 times. For two-stroke LSEs, the boost pressure is 0.35...0.4 MPa, which corresponds to an average effective pressure of 1.6...2.1 MPa.

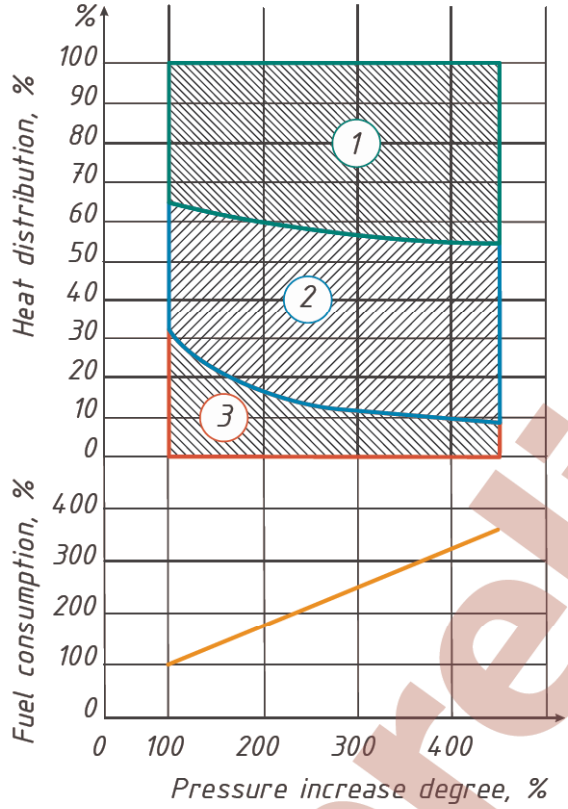


Figure 4.49 – Redistribution of the components of the thermal balance of engines as a function of the degree of boost: 1 – heat converted into mechanical energy; 2 – heat transferred to the cooling media; 3 – heat loss with exhaust gases (adapted from [40])

In Fig. 4.50 *a* shows the general diagram of supercharging a four-stroke engine using a turbocharger unit with a constant pressure turbine and a pulse turbine (Fig. 4.50 *b*).

Along with increasing the air pressure at the inlet, to increase the weight charge, charge air is cooled in special heat exchange units – air coolers. In turn, cooling the air can lead to condensation and loss of moisture contained in it. To remove it, special devices are installed – moisture separators.

Thus, the air-gas system of a modern marine diesel engine includes turbocharger units, charge air coolers, moisture separators, receivers and connecting elements.

Modern marine diesel engines use gas turbine charging systems, which can be classified according to the following criteria:

- according to the number of air compression stages, single-stage and two-stage. In a single-stage system, air is compressed in a single-stage turbocharger (Fig. 4.48, 4.50). In a two-stage system, air is compressed sequentially, moving from the first compression stage to the second (Fig. 4.51 *a*);
- according to the number of turbochargers and the method of their connection – single-compressor and multi-compressor. In single-compressor systems, all cylinders are pressurized from one compressor (Fig. 4.48 *c*, 4.50). In multi-compressor systems, compressors can be connected in series (with two-stage charging) (Fig. 4.51 *a*) or in parallel (with single-stage charging) (Fig. 4.51 *b*). A number of designs provide for the shutdown of part of the turbochargers when switching to partial load modes. If the charging system uses more than two turbochargers, their connection may be different;

For preliminary review

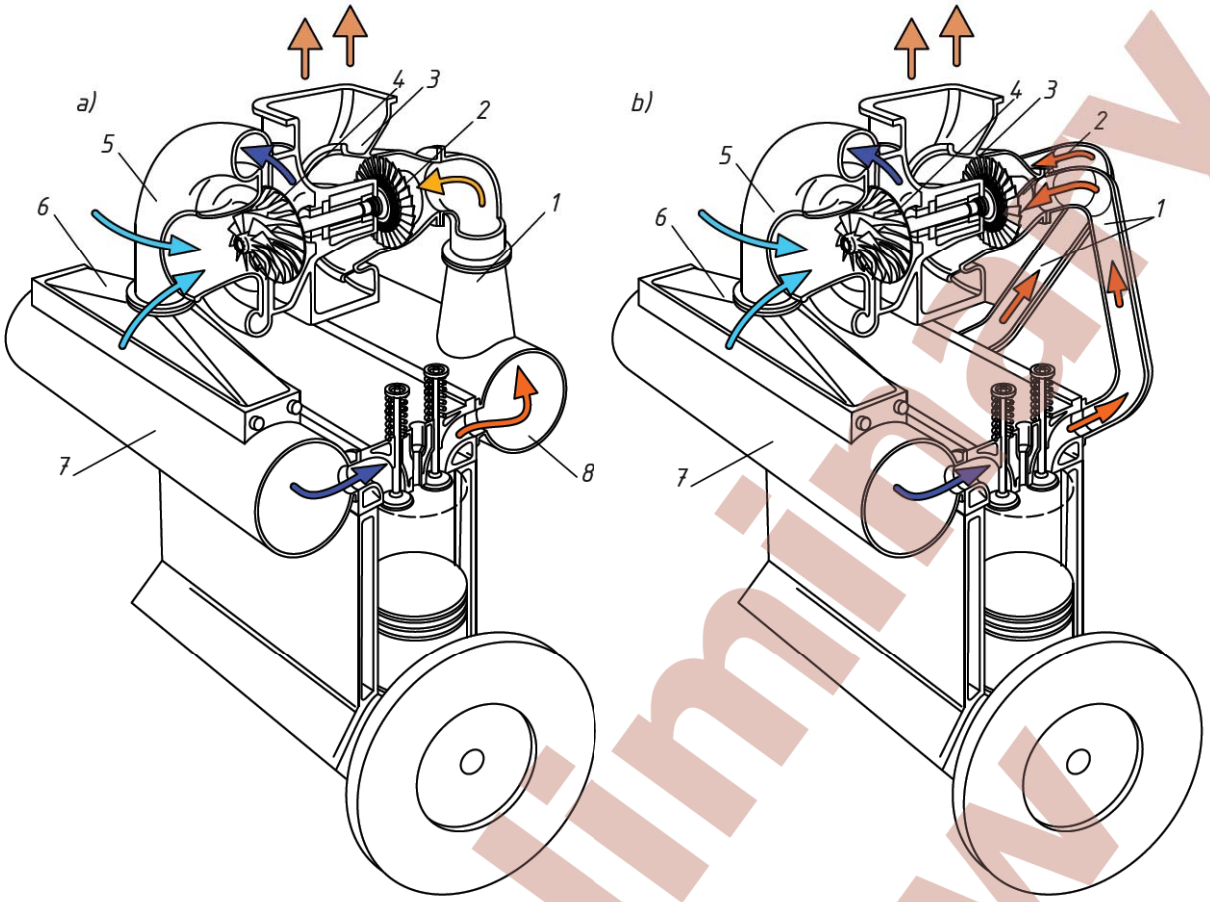


Figure 4.50 – General diagram of supercharging a four-stroke engine using a turbocharger unit with a constant pressure turbine (*a*) and a pulse turbine (*b*): 1 – gas supply pipe to the turbine; 2 – axial turbine wheel; 3 – flue for exhaust gases; 4 – centrifugal compressor wheel; 5 – air supply pipe from the compressor; 6 – air cooler; 7 – inlet receiver; 8 – exhaust receiver (adapted from [2, 39])

- according to the method of supplying exhaust gases to the turbine – in the form of pressure pulses (pulse boost) (Fig. 4.50 *a*) and at constant pressure (isobaric boost) (Fig. 4.50 *b*);
- by the method of maintaining the parameters of the air entering the cylinders: *unregulated,* in which the parameters of the charge air change depending on the operating mode of the engine or external conditions, and *adjustable,* in which the value of any of the state parameters is maintained at a given level or changes according to a predetermined law air (temperature, pressure, flow);
- according to the number and method of connecting heat exchangers – with air coolers and heaters.

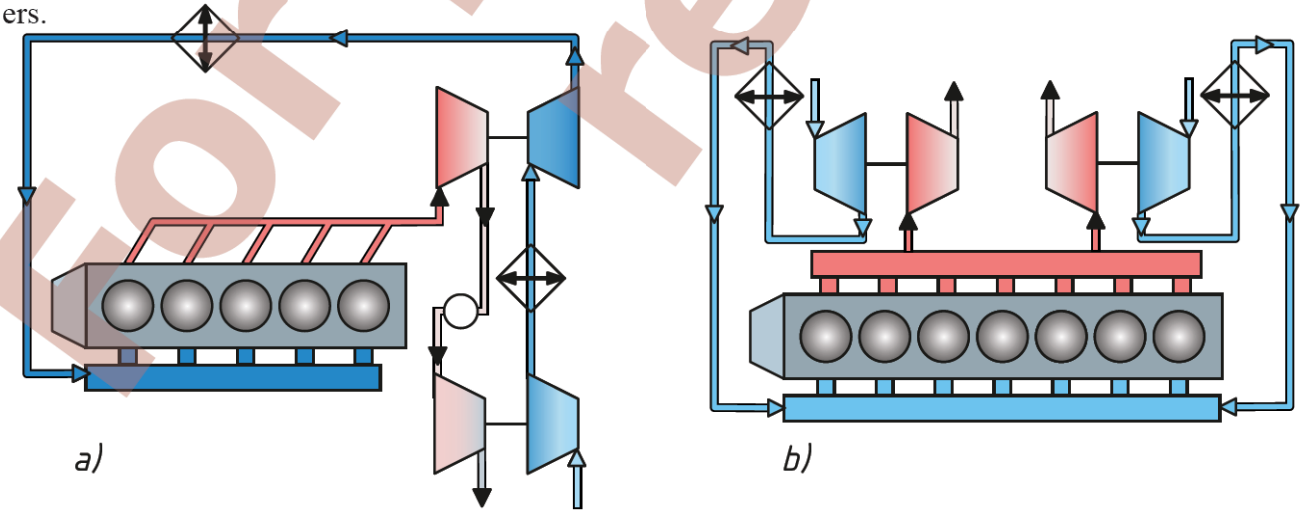


Figure 4.51 – Supercharging using two turbochargers: *a* – two-stage sequential pulse charging with two-stage cooling; *b* – parallel isobaric pressurization with end cooling. ◀ – exhaust gases; ◁ – air

For preliminary review

Modern diesel engines are equipped with twin heat exchangers, one of which is connected to the hot water circuit and the other to the cold water circuit. By redistributing the flows between two heat exchangers, it is possible to cool or heat the air, maintaining a stable temperature at the inlet to the working cylinders. If the cooler is installed directly in front of the intake receiver, it is called *an end cooler*. A heat exchanger installed between compressors connected in series is called *intermediate*. A system with intermediate and end heat exchangers is called a two-stage charge air cooling system.

4.7.4 Turbocharger units for supercharging marine diesel engines

Marine diesel turbochargers are units in which the compressor and turbine wheels are mounted on the same shaft (Fig. 4.52). If gases are supplied to the turbine blades and removed from them in a plane coinciding with the axis of rotation, then such a turbine is called axial (Fig. 4.52 *a*). If gases are supplied to the turbine blades in a plane coinciding with the axis of rotation and removed in a radial plane, then such a turbine is called radial-axial (Fig. 4.52 *b*).

Both types of turbines are used in turbochargers of modern marine diesel engines. At relatively low gas flow rates, preference is given to radial-axial turbines; at high gas flow rates, preference is given to axial turbines. This is explained by the fact that at low volumetric gas flow rates, the blades of the axial turbine have a small height, which leads to an increase in losses in the stage and a decrease in its efficiency. Therefore, the efficiency of a radial-axial turbine is slightly higher than the efficiency of an axial turbine. With an increase in gas flow through the turbine, the height of the blades increases, the relative value of losses decreases, and the efficiency of an axial turbine becomes higher than that of a radial-axial one.

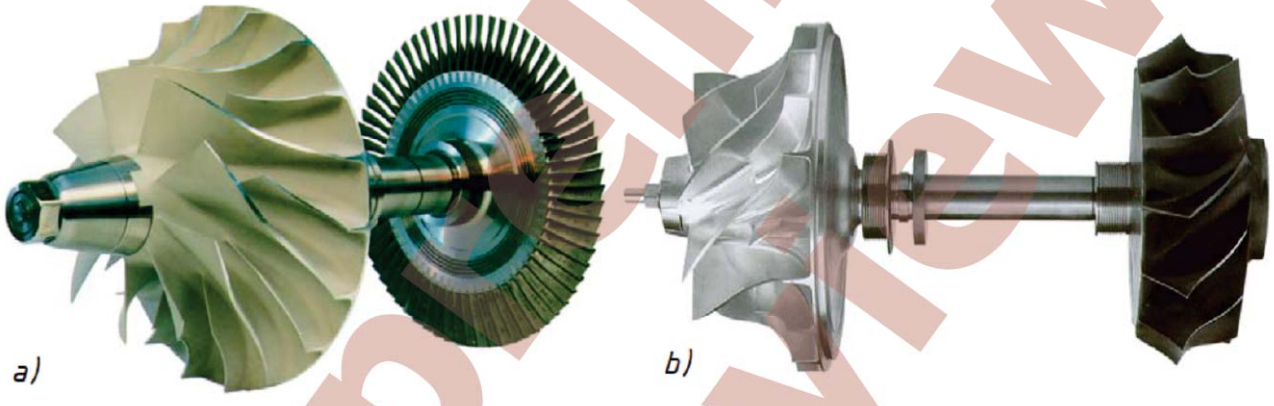


Figure 4.52 – Rotors of turbochargers with axial (*a*) and radial-axial turbine (*b*) (adapted from [42, 43])

The compressors in all modern turbo units are centrifugal, despite the fact that their efficiency is less than that of axial ones, and their productivity is greater. This is explained by the fact that the degree of pressure increase in the stage of a subsonic axial compressor is 1.07...1.1. Thus, to obtain a boost pressure of 0.25 MPa, it is necessary to install up to 12...14 stages. At the same time, such boost pressure is easily achieved in a single-stage centrifugal compressor. In addition, in a centrifugal compressor, clogging of the flow path has less effect on air flow, the degree of pressure increase and efficiency. A centrifugal compressor is simpler and cheaper than an axial compressor.

For the reasons given above, turbochargers consisting of a single-stage centrifugal compressor and a single-stage axial or radial-axial type turbine are currently used to supercharge marine engines.

Next, the general structure and operating principle of individual units included in turbochargers of various types and purposes will be briefly discussed.

4.7.4.1 General structure and principle of operation of a radial-axial centrifugal compressor. *A centrifugal compressor* (Fig. 4.53 *a*) includes an inlet device, an impeller (also called an impeller), a diffuser consisting of a bladeless and bladed part, and an air collector, often made in the form of a spiral (scroll).

For preliminary review

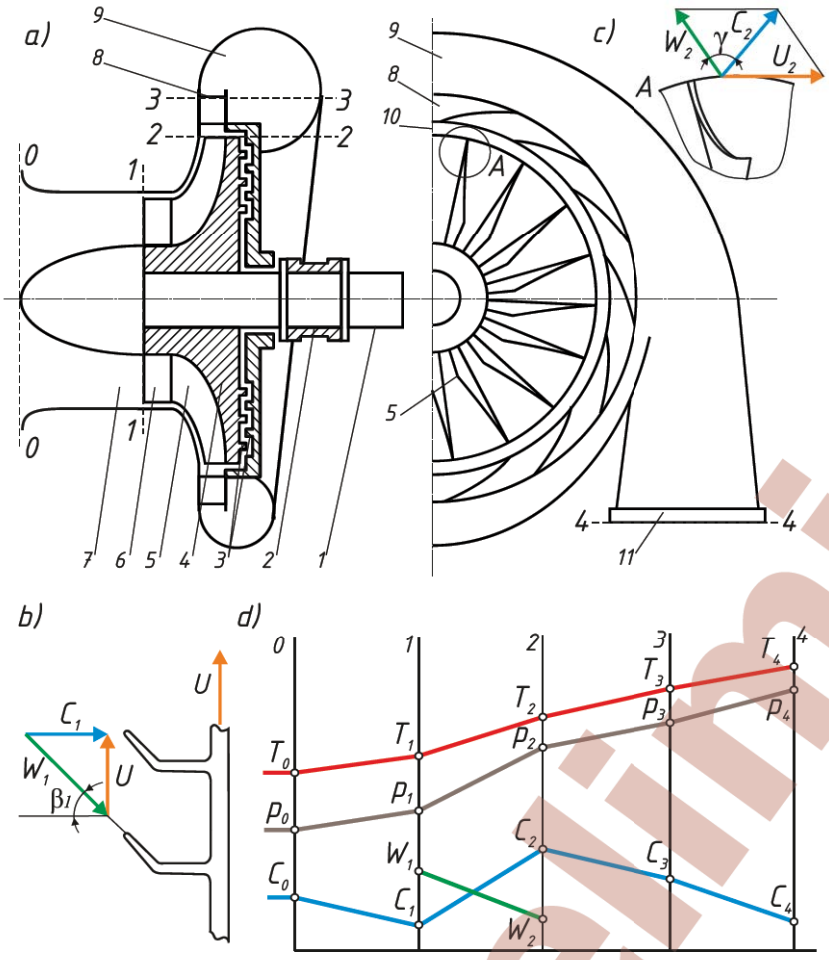


Figure 4.53 – Diagram of a radial-axial centrifugal compressor (*a*), speed distribution at the inlet to the impeller (*b*) and at its outlet (*c*), as well as changes in gas parameters in the main sections of the compressor (*d*): 1 – drive shaft; 2 – bearing; 3 – labyrinth seal; 4 – impeller hub; 5 – impeller blades; 6 – rotating guide vane; 7 – input device; 8 – diffuser blades; 9 – snail-type air collector; 10 – bladeless section of the diffuser; 11 – outlet pipe

The air, having passed through the filter, enters the inlet device, which serves to increase the stability of the flow and is tapered. Thus, the inlet device ensures a uniform supply of air to the wheel with minimal losses. The compressor impeller is mounted on the same shaft as the gas turbine impeller. It is a disk with end radial blades forming expanding interscapular channels. The diffuser is made in the form of an expanding channel located between the walls of the housing. It has a bladeless section, which is an annular gap between the end of the impeller and the diffuser blades, and a blade section formed by a system of blades located in a circle around the compressor wheel, with expanding channels in the direction of air movement. On small turbochargers, the blade section of the diffuser may be missing. The outlet pipes are expanding channels located in a circle around the diffuser. They collect the air leaving the diffuser and direct it into the engine intake pipes. Often the outlet pipes are made in the form of a so-called «snail».

To explain the principle of operation of a centrifugal compressor, let us consider the movement of air in its elements, identifying the most characteristic sections (Fig. 4.53 *a*): *0-0* – cross section at the inlet to the compressor. Air parameters in this section: pressure p_0, temperature T_0 and speed c_0; *1-1* – cross section at the entrance to the impeller (p_1, T_1, c_1); *2-2* – exit from the impeller (p_2, T_2, c_2); *3-3* – outlet from the diffuser with air parameters p_3, T_3, c_3; *4-4* – output from the compressor with air flow parameters p_4, T_4, c_4.

When a centrifugal compressor operates, a vacuum is formed at the entrance to the impeller, under the influence of which air (cross section *0-0*) with parameters p_0, T_0, c_0 enters the input device.

Air enters the impeller with an absolute speed c_1, pressure p_1 and temperature T_1. Rotating, the compressor wheel involves the air entering it into rotational motion (section 1-1), with a peripheral (transferable) speed U_1 (Fig. 4.53 *b*). Next, the flow moves in the inter-blade channels of the wheel at a speed W_1, called the relative speed, equal to the geometric difference between the absolute speed c_1 and the circumferential speed U_1. In order to ensure a smooth (impact-free) entry of air into the impeller, it is necessary that the relative velocity vector W_1 coincides with the direction of the leading edges of the wheel blades (angle β_1). The leading edges of the wheel blades, bent to a cer-

For preliminary review

tain angle, form a rotating (moving) guide vane.

Further, the air, under the influence of centrifugal forces of inertia, is thrown to the periphery and, passing through the expanding inter-blade channels of the impeller, is compressed due to the conversion of part of the kinetic energy of the flow into potential energy, while the pressure (p_2) and temperature (T_2) of the air increase (section 2-2). As the air moves through the channels, it accelerates and at the exit from the wheel the flow speed reaches a value of c_2. The magnitude of this speed is determined as the sum of the vectors of the peripheral (transferable) speed U_2, directed tangentially to the circumference of the wheel, and the relative speed W_2, deviated from the radial direction by an angle γ (Fig. 4.53 *c*) due to the uneven distribution of pressures and velocities inside channel.

The kinetic energy at the wheel exit is usually about half of the total flow energy, so to convert it into pressure energy, a diffuser is installed behind the impeller. When air moves in the diffuser, due to a continuous increase in the cross-sectional area, the flow velocity drops (c_3), as a result of which it slows down and the kinetic energy transforms into potential energy. This leads to a further increase in pressure (p_3) and an increase in temperature (T_3). Losses arising during air movement in the diffuser constitute a significant proportion of the total losses in the compressor; to reduce them, blades are installed that form separate expanding channels.

The increase in air pressure to p_4 continues in the expanding channels of the outlet pipes due to the deceleration of the flow to a speed c_4. In this case, the air temperature increases (T_4).

Changes in air parameters (pressure p, temperature T and speed c) during its movement in the compressor elements are presented graphically in Fig. 4.53 *d*. The numbers on the graph indicate characteristic sections. The numbers on the graph and on the compressor diagram (Fig. 4.53 *a*) indicate the same sections.

4.5.4.2 The general structure and principle of operation of the axial gas turbine stage are shown in Fig. 4.54 *a*. A general view of a turbocharger with an axial single-stage turbine is shown in Fig. 4.55.

The main elements of the turbine stage are a fixed inlet guide vane and an impeller with blades mounted on it. The blades of the guide vane and the turbine impeller have special aerodynamic profiles and form a system of channels for the flow part of the turbine, through which the flow of gas supplied from the engine flows.

To consider the principle of operation of an axial turbine stage, let us depict its working bodies in the form of a turbine array, which represents the development of the blades of the inlet guide vane and the rotor onto a plane along the section line *a-a* (Fig. 4.54 *b*).

Gases from the exhaust receiver or engine manifold are supplied to the inlet guide vane with an absolute flow rate c_1, pressure p_1 and temperature T_1. Next, the gases enter the channels of the guide vane apparatus, which are made tapering, that is, the cross-section of the channels f at the outlet is less than at the inlet ($f_{1'} < f_1$). This leads to the fact that when moving through the channels, the potential energy of the pressure of the gas flow turns into kinetic energy, and the flow speed increases c_1 to $c_{1'}$, at the same time the pressure and temperature of the gas drop ($p_{1'} < p_1$ and $T_{1'} < T_1$).

Coming out of the inlet guide vane, the gases, with an absolute speed $c_{1'}$, enter the blades of the impeller rotating at a peripheral speed U. In the inter-blade channel formed by the blades of the impeller, gases move with a relative speed $W_{1'}$, which at the exit from the channels is equal to the geometric difference between the absolute $c_{1'}$ and circumferential speed U, that is, $W_{1'} = c_{1'} - U$.

To ensure shock-free entry, the leading edges of the impeller blades are installed in the direction coinciding with the direction of the relative velocity vector $W_{1'}$. Due to the increase in peripheral speed from the base of the blade to the tip and the need to ensure shock-free entry at all radii, the impeller blade has a variable profile in height, with rotation relative to the base (blade twist).

At the impeller, the kinetic energy of the gas flow is converted into mechanical work. The absolute flow velocity decreases in the impeller channels $c_{1'}$ to c_2.

Depending on the type of turbine, the gas in the inter-blade channels of the impeller either continues to expand (pressure drops from $p_{1'}$ to p_2), or only changes the direction of movement, and the pressure remains unchanged.

For preliminary review

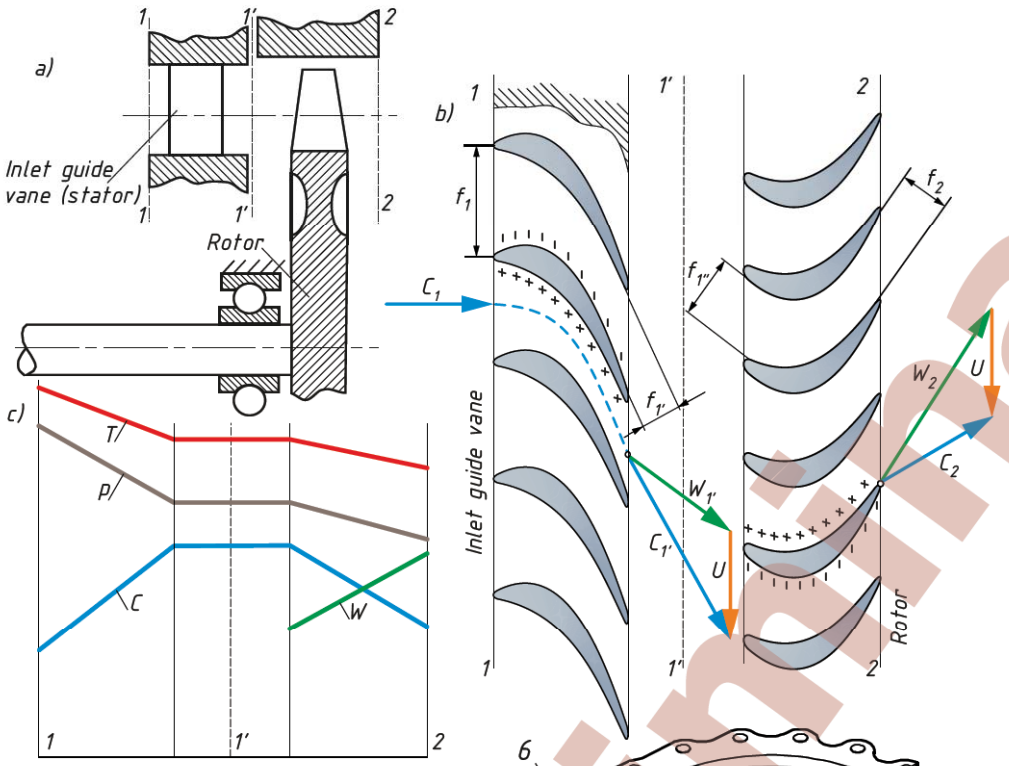


Figure 4.54 – Stage diagram (*a*) and turbine array of an axial gas turbine (*b*), change in gas parameters in the stage of an axial turbine (*c*)

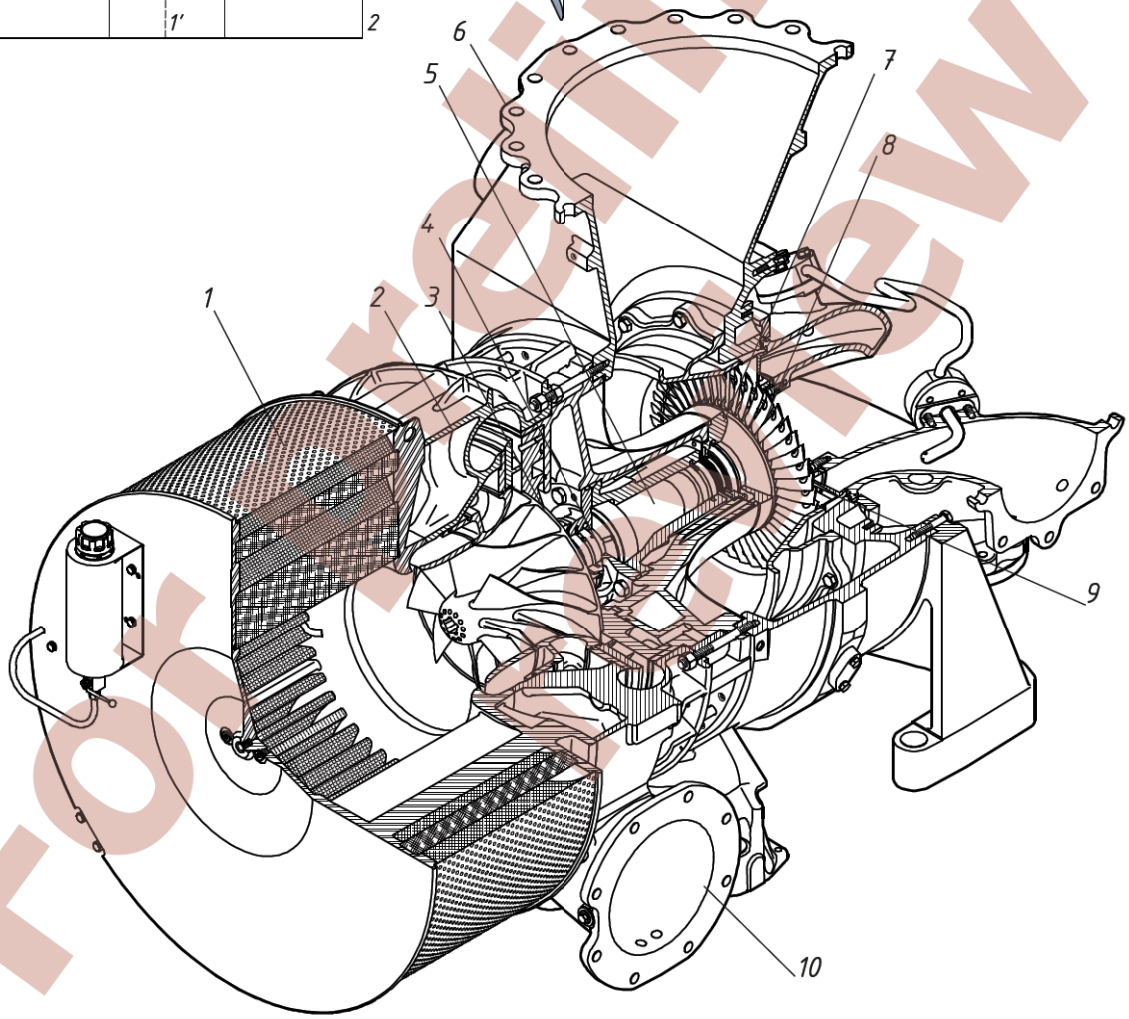


Figure 4.55 – Turbocharger with an axial turbine type TPL76-C from ABB: 1 – air filter; 2 – compressor guide vane; 3 – compressor wheel; 4 – rotor shaft bearing; 5 – rotor shaft; 6 – gas supply pipe to the turbine; 7 – seal of the turbine bearing unit; 8 – turbine wheel; 9 – turbine guide vane; 10 – air exhaust pipe (adapted from [44])

For review

Turbines of the first type are called reactive, and the second type are called active. In the practice of designing turbochargers for supercharging marine internal combustion engines, jet turbines are most widely used, since they have a higher efficiency in the design mode and are less sensitive to manufacturing inaccuracies.

In a jet turbine, the inter-blade channels are made tapering ($f_2 < f_{1''}$). In this regard, the relative gas speed in the channels of the impeller of such a turbine increases from the value $W_{1'}$ to W_2. Mechanical work on the shaft of the jet turbine is performed due to the impact on blade of the active impulse of the gas flow and due to the reactive force that arises when the gas jet accelerates from the speed $W_{1'}$ to W_2.

The disadvantage of axial jet turbines is the action of significant axial forces and higher peripheral speeds of the wheel, as well as higher stresses in the blades and disk.

4.7.4.3 General structure and principle of operation of the radial-axial gas turbine stage. In radial-axial turbines, gas moves radially from the periphery to the center and, after making a 90° turn, exits the turbine in the axial direction. Otherwise, the operating principle of radial-axial turbines does not differ from axial turbines. The diagram of a turbine stage of this type and the speed distribution in it is shown in Fig. 4.56.

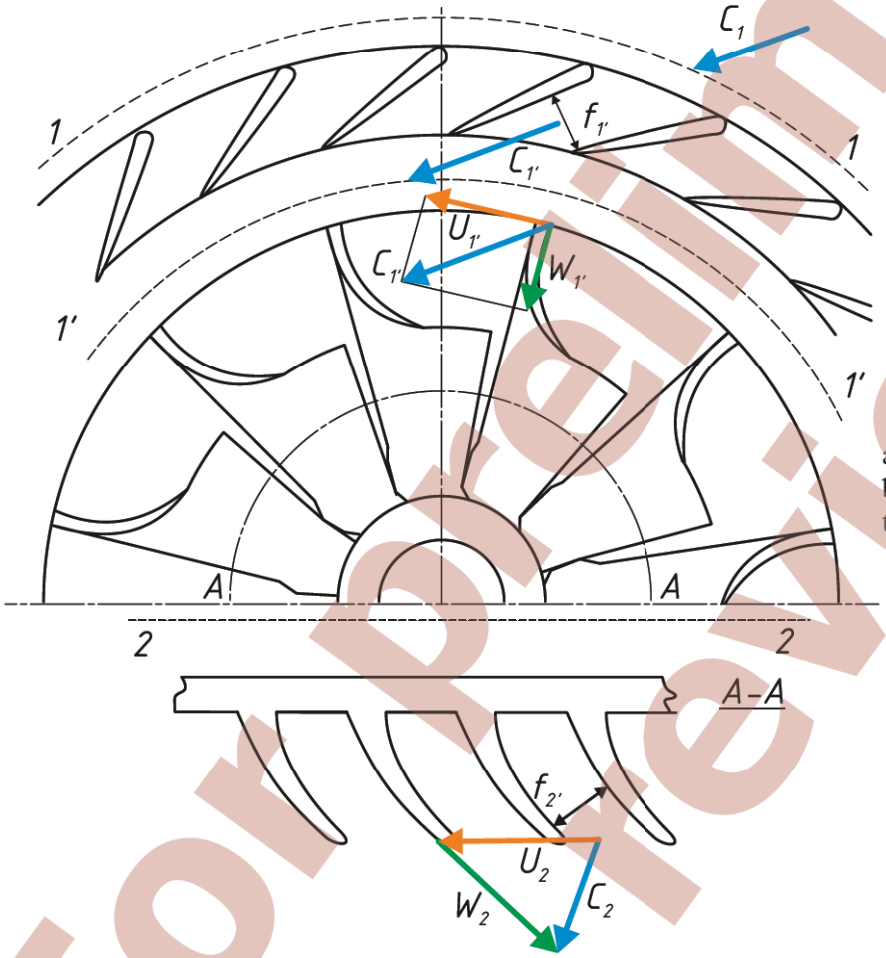


Figure 4.56 – Stage diagram and speed distribution in the turbine cascade of a radial-axial gas turbine

Unlike axial turbines, which can be either active or reactive, radial-axial turbines are made only reactive. This is explained by the need to overcome centrifugal forces when gas moves from the periphery in the radial direction. For small turbines, a bladeless guide vane is used, which significantly simplifies the design.

Compared to axial turbines, radial-axial turbines are distinguished by the ease of manufacturing of impellers, their increased strength, and less sensitivity to violations of the geometry of the flow path.

The disadvantages of radial-axial turbines are the increased transverse size of the casing and the high number of rotor revolutions.

A general view of a turbocharger with a radial-axial turbine is shown in Fig. 4.57.

For preliminary review

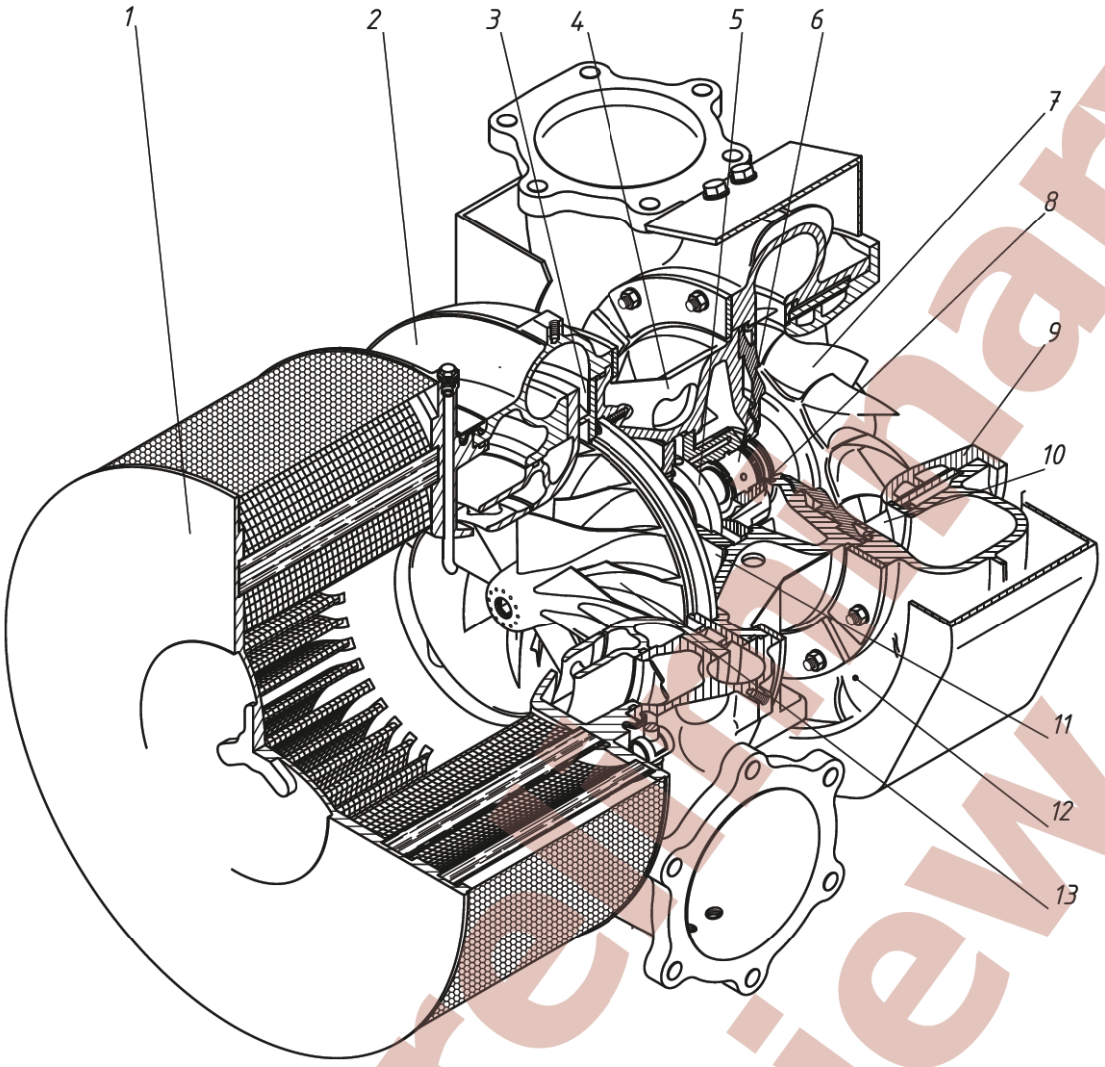


Figure 4.57 – Turbocharger with radial turbine type TPS52-F from ABB: 1 – air filter; 2 – compressor outlet; 3 – diffuser; 4 – bearing housing; 5 – axial bearing; 6 – radial bearing on the side of the turbine wheel; 7 – turbine wheel; 8 – bearing unit; 9 – turbine deflector; 10 – turbine guide vane; 11 – seal on the compressor side; 12 – turbine outlet pipe; 13 – compressor wheel (adapted from [45])

4.7.4.4 Design features of individual elements of turbocharger units. All turbochargers consist of fixed elements and a movable rotor, which in turn consists of turbine and compressor wheels, as well as the shaft connecting them. As a rule, the selected rotor design determines the design of the entire turbocharger unit as a whole, including housing elements. In modern marine internal combustion engines find use of rotor designs in which the turbine and compressor wheels are located at different ends of the shaft connecting them. In this case, the bearings on which the shaft rests can be located both along its edges and between the impellers.

The most commonly used rotor designs are shown in Fig. 4.58.

For turbochargers made according to the scheme shown in Fig. 4.58 *a*, bearings and seal assemblies are located at the edges of the rotor (the so-called cantileverless design). With this arrangement, the bearings on the turbine side are the furthest from the disk and are not exposed to high temperatures.

Bearings are usually mounted in separate housings at the edges of the unit, so they are easily accessible for lubrication, maintenance and replacement. The bearings used have a small diameter, so their peripheral speeds are relatively low, which, in turn, reduces heat generation and reduces the likelihood of overheating. In such designs, along with plain bearings, rolling bearings are often used to reduce mechanical losses in the unit.

For preliminary review

The factors listed above determine the high reliability, long service life and ease of use of units of this type.

The disadvantages of the scheme under consideration include:

- increased dimensions and weight;
- due to the larger distance between the bearings, the shaft must be provided with greater rigidity. To do this, it is made with a larger diameter, which increases the inertia of the rotor and increases its acceleration or braking time when the engine load changes;
- since the bearing seats are located in different housing parts, it is difficult to ensure their alignment during manufacturing and assembly;
- uneven expansion of the housing due to uneven temperature distribution can also lead to bearing misalignment. To reduce the influence of temperature differences, the casings of turbochargers of this type are made water-cooled ;
- due to the lateral location of the supports, it is more difficult to profile the inlet devices of the compressor and turbine;
- worse access to the compressor and turbine wheels.

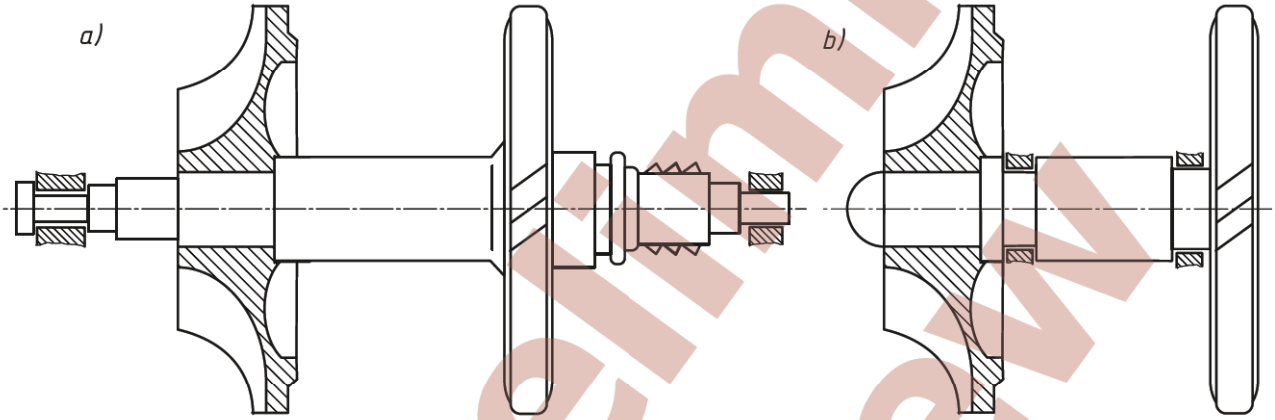


Figure 4.58 – Schemes for installing turbocharger rotors of modern marine internal combustion engines: *a* – with bearings located at the edges of the rotor (non-cantilever scheme); *b* – two-console design with supports located between the compressor and turbine disks (adapted from [43])

The most striking representative of the class of turbochargers made according to the first scheme are units of the VTR series (Fig. 4.59 *a*) from ABB, which inherited the developments of Brown Boveri. It consists of a single-stage reaction-type gas turbine and a centrifugal compressor. The rotor is solid forged and rotates in two bearings located at diametrically opposite ends of the rotor. One of the bearings on the turbine side is a roller or ball support, single row. Both bearings have elastic outer races that protect them from the effects of housing vibration. Bearing lubrication oil is poured into the bearing housings located at the turbine and compressor ends of the rotor.

When creating turbochargers of the VTR series of large sizes, due to the large diameter of the rotor, the company uses plain bearings.

The two-console circuit shown in Fig. 4.58 *b*, with supports located between the compressor and turbine disks, ensures small overall dimensions and weight of the turbocharger, which often facilitates its location on the engine. The close arrangement of bearings from each other in one housing part almost completely eliminates the possibility of their alignment being disrupted due to uneven temperature distribution. In this regard, turbochargers of this type can operate without water cooling of the housing at all, or with cooling only in the area where the bearing is installed on the turbine wheel side. The absence of cooling of the gas supply channels increases the efficiency of the turbocharger and reduces the likelihood of corrosion of its housing when operating on heavy fuels. The absence of bearing units at the ends of the housing simplifies the profiling of gas channels. They can be performed with slight turns, or even straight lines. The small mass of the rotor ensures its lower inertia and faster response of the turbocharger to changes in engine load.

For Preliminary review

The main disadvantage of the scheme is the impossibility of inspecting the bearings without disassembling the rotor. Larger diameter bearings imply higher friction within them, which results in increased heat generation. Due to the intense heating of bearings, their effective cooling is necessary, especially on the turbine side. In addition, high demands are placed on the material, design and quality of manufacture of bearings, and the smaller distance between the rotor supports requires more accurate balancing.

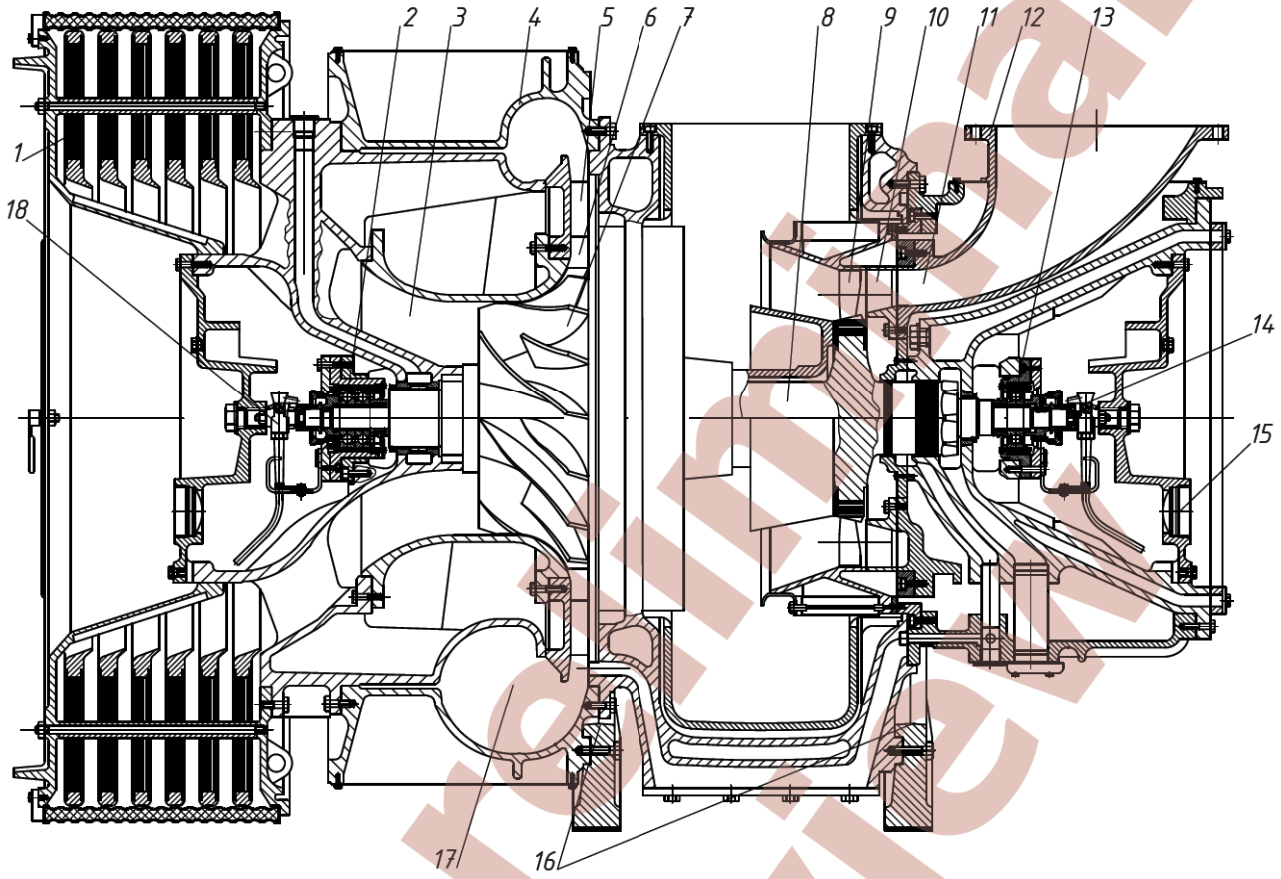


Figure 4.59 – Turbocharger type VTR-454 from ABB: 1 – filter muffler; 2 – bearing unit; 3 – air chamber; 4 – compressor housing; 5 – blade diffuser; 6 – bladeless diffuser; 7 – compressor impeller; 8 – rotor; 9 – turbine impeller; 10 – guide vane; 11 – gas supply chamber; 12 – gas inlet pipe; 13 – bearing unit; 14 – oil pump of the bearing assembly on the compressor side; 15 – sight glass; 16 – mounting feet; 17 – snail; 18 – oil pump of the bearing assembly on the turbine side (adapted from [46])

Due to its simplicity, this scheme is becoming increasingly widespread. It is used to manufacture the bulk of turbochargers newly developed for marine internal combustion engines.

Until recently, the largest manufacturers of turbochargers of this design were MAN and Mitsubishi, but in 1999 the Swiss company ABB, which had been producing turbochargers using a consoleless design for a long time, developed a new model range of turbochargers such as TPL (Fig. 4.55) and TPS (Fig. 4.58) in a two-console design.

4.7.4.5 Design of rotor impellers and inlet guide vane devices

The turbine rotor wheel consists of a hub made of alloy steel, which is welded to the shaft using friction welding. The blades, made of heat -resistant alloy steel, are attached to the wheel hub using shanks with a special profiled herringbone notch (Fig. 4.60, 4.61 *a*). These connections have a large supporting surface and a significant area of the most loaded section of the blade shank.

The herringbone shank provides the smallest radial dimension of the disc rim and minimal stresses compared to other types of shanks.

For preliminary review

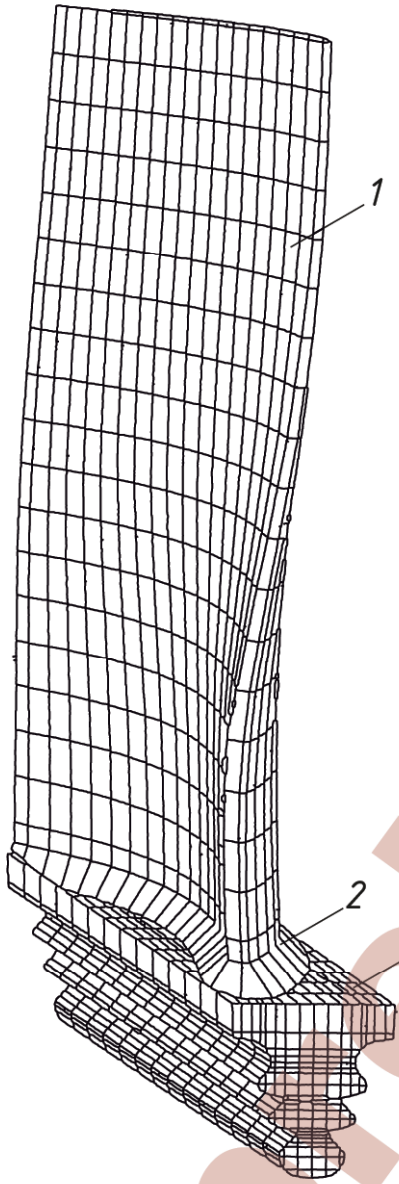


Figure 4.60 – Turbocharger turbine blade: 1 – blade feather; 2 – transition fillet; 3 – herringbone shank (adapted from [47])

The centrifugal compressor wheel is usually made from one piece of aluminium alloy by high-precision injection molding or milling. The latter method provides greater manufacturing precision and, therefore, higher compressor efficiency. On turbochargers of modern MICEs, wheels of the so-called semi-open type are used, that is, the working channels form the blades of the wheel, its disk and the walls of the stationary chamber of the compressor.

The compressor wheel is connected to the rotor using a keyed or splined connection and secured with a nut at the end of the shaft (Fig. 4.61 *b*). On smaller turbochargers, the compressor wheel may be mounted to the shaft using a press fit.

On large turbochargers, the rotating guide vane can be made in the form of a separate wheel, which is mounted motionlessly on the shaft or attached to the compressor wheel using threaded connections.

The turbine and compressor are separated from the cavities of the thrust and support bearings by labyrinth seals formed by scallops rolled into the housing and protrusions made on the back side of the wheels (Fig. 4.61 *c*). Similar seals are installed on the rotor shaft on both the compressor and turbine sides.

Air with excess pressure is supplied to the labyrinth seals through drillings in the housing, which prevents the occurrence of vacuum in the bearing cavity and oil carryover into the compressor. Through drillings in the middle housing and the turbine housing, air also enters the cavity between the turbine wheel and the labyrinths, which helps cool the rotor and prevents hot gases from breaking through into the bearing cavity.

For preliminary review

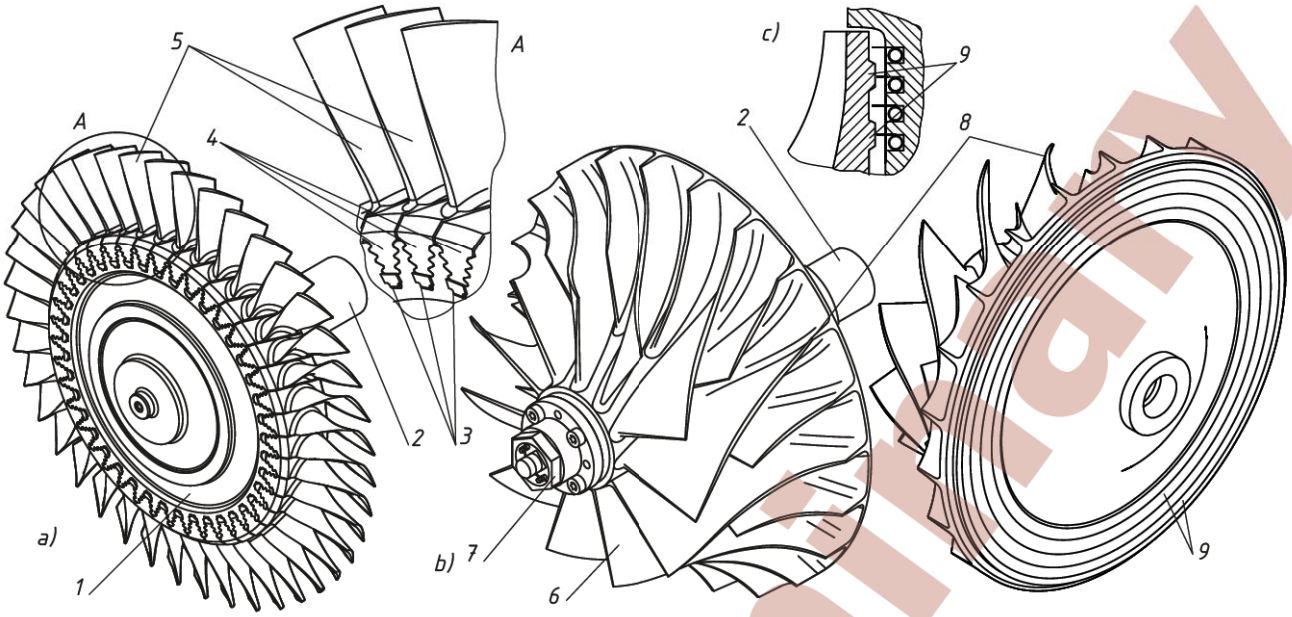


Figure 4.61 – Turbine wheel (*a*) and compressor wheel (*b*) of a TCA type turbocharger from MAN. Labyrinth seal of the compressor wheel (*c*): 1 – turbine wheel hub; 2 – rotor shaft; 3 – locking strips for fastening the blades; 4 – fastening of the blade shank of the «christmas tree» type; 5 – turbine blades; 6 – compressor blades; 7 – compressor wheel fastening nut; 8 – compressor wheel hub; 9 – protrusions of the labyrinth seal (adapted from [48])

The turbine inlet guide vane plays an important role in shaping the characteristics of both the turbine and the entire turbocharger unit as a whole. In the inlet guide vane, an initial increase in the speed of the gas flow occurs and the angle of its entry onto the turbine wheel blades is formed. As has already been shown above, deviation from this angle significantly increases energy losses from the impact of the gas flow on the turbine blades, as a result of which the efficiency of the turbine drops sharply.

The turbines of modern MICE supercharging units use guide vane devices with both fixed and movable blades.

Devices with fixed guide vane arrays are simpler in design however they provide high turbine efficiency only in a very narrow range of operating conditions. Structurally, a fixed inlet guide vane usually consists of outer and inner rings, between which blades are, installed (Fig. 4.62).

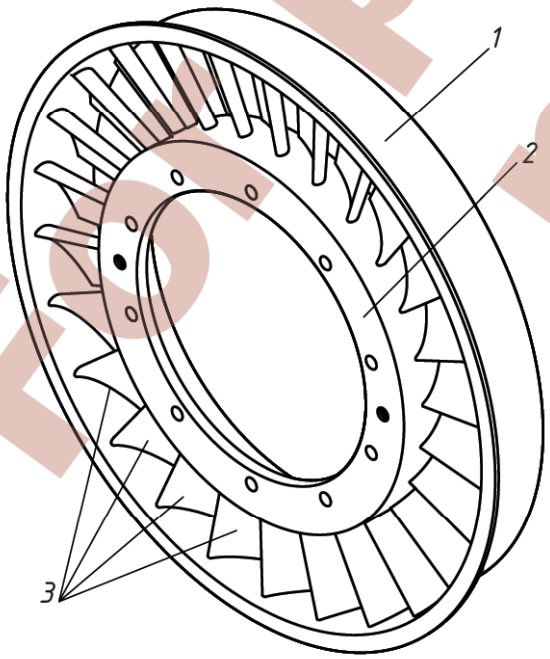


Figure 4.62 – Inlet guide vane of an axial turbine of a TCA type turbocharger from MAN with a fixed blade grid: 1 – outer ring; 2 – inner ring; 3 – blades (adapted from [48])

For preliminary review

The blades, stamped from austenitic steels, can be attached to the rings using spikes or mounted in special molds and filled with cast iron, forming the inner and outer rings of the guide vane. Compensatory cuts are made around the circumference of the diaphragm, running from the center of the inner ring to the blades. The inlet guide vane is usually attached to the turbine housing with bolts. In order to harmonize the modes of joint operation of a turbocharger and a diesel engine, most manufacturers produce for each standard size of units several types of guide vane devices, differing in blade angles and inlet areas. To better match the characteristics of the turbocharger with the characteristics of the engine over a wide range of loads, guide vane devices with variable geometry are used.

The diffuser blade apparatus of radial-axial compressors can also be made either fixed or with variable channel geometry. Structurally, fixed compressor devices are made in the form of a ring with blades located on the end surface (Fig. 4.63).

The rings are installed in the groove of the housing, and the blades, pressed with their free end to the surface of the air intake casing, form the diffuser channels.

Diffusers with variable geometry of radial-axial compressors do not differ in design from the inlet guide vane devices of radial-axial turbines.

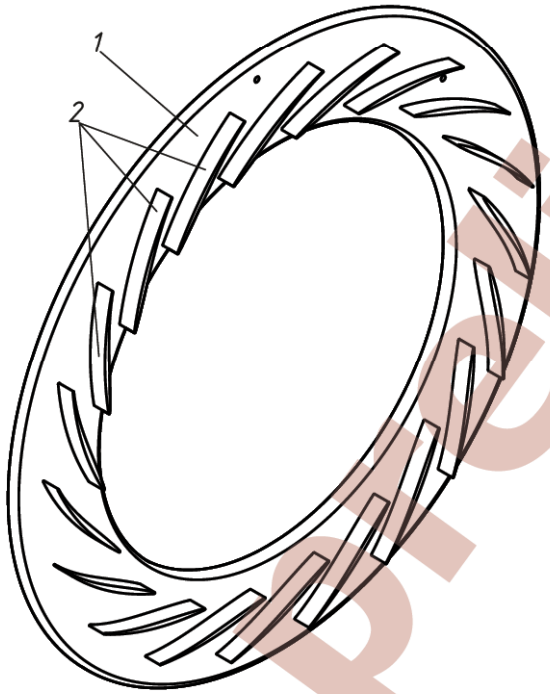


Figure 4.63 – Vane diffuser of a turbocharger compressor type TCA from MAN: 1 – ring; 2 – blades (adapted from [48])

In some designs, to optimize the angle of gas entry onto the compressor wheel blades, an adjustable guide vane is installed in front of it in the inlet device. The presence of this device allows you to optimize the angle of gas entry onto the impeller blades, reducing the likelihood of the so-called «surge», a phenomenon associated with the appearance of pressure waves in the compressor channels arising from pressure differences on different sides of the blades. In this case, flow disruptions are observed in individual channels of the compressor wheel. The operation of the compressor is accompanied by intense noise due to fluctuations in suction pressure. Rotor vibration occurs, which can lead to damage to the blades and bearings.

4.7.4.6 Design of bearings and bearing units

Rolling and plain bearings are used in turbochargers. The choice of bearing type is determined by the design features of the turbocharger and the requirements for it.

Rolling bearings have lower mechanical losses and provide good starting conditions for the turbocharger. The disadvantages of these bearings include high cost, relatively short service life, and the need for individual lubrication, which complicates the design of the bearing assembly.

Rolling bearings are used mainly when bearing units are located at the ends of the rotor, when it is relatively easy to obtain small diameters of the shaft journals. With a cantilever installation of the rotor, the diameter of the shaft bearing journals increases, and it becomes difficult to ensure the

For preliminary review

functionality of the rolling bearings.

To reduce dynamic loads, rolling bearings are installed on elastic damping supports.

In Fig. 4.64 *a*, *b* shows the design of bearing units with elastic support from ABB. Elastic support elements perceive dynamic, vibration loads and unbalanced variable forces arising from imbalance of the rotor. On the compressor side, they are made in the form of a package of concentrically located rings made of spring steel, in which holes are made, and special rubber liners are installed between the plates. On the side of the turbine, where the temperature is higher, steel rings are used, which have protrusions stamped on the outer and inner surfaces that alternately rest on both rings. During the operation of the support, the oil located between the steel plates and rubber liners serves as a hydraulic vibration absorber, in which the energy of shocks and vibrations is spent on displacing oil through narrow cracks and holes in the steel plates.

Bearing units are equipped with an individual lubrication system. Oil is supplied by disks that lift it into the upper cavity, from which it flows by gravity into the bearings (Fig. 4. 64 *a*). To supply lubricant, radial drillings are made in the disk, through which the oil flows out under the influence of centrifugal force and is distributed over the inner surface of the catching casing. As a result, a vacuum is created in the inner chamber of the disk, which is transmitted through a sealing system to the suction tube. Under the influence of this vacuum, oil is sucked into the tube from the lubricant cavity of the bearing assembly.

In large turbochargers, oil is supplied to the bearings by built-in oil pumps driven by the rotor (Fig. 4.64 *b*). For lubrication, low-viscosity oil is used, which is stored in the internal cavity of the bearing assembly. Bearings are replaced without disassembling the turbocharger, which significantly reduces repair costs. The service life of rolling bearings, according to ABB, ranges from 6 to 16 thousand hours, depending on the type of compressor and its operating conditions.

Sliding bearings are structurally much simpler and cheaper, have greater durability and can be lubricated with oil from the engine lubrication system. Bearings of this type have low specific loads and high peripheral speeds, which allows for complete hydrodynamic lubrication in the main operating modes. This in turn leads to a significant reduction in wear.

Sliding bearings of turbochargers of modern marine internal combustion engines require virtually no maintenance and provide resources commensurate with the resources before a major overhaul of the engines themselves on which they are installed. These qualities have ensured the predominant spread of plain bearings in newly created designs of turbochargers for marine use.

Structurally, plain bearings for turbochargers can be made in the form of bushings rigidly pressed into the housing of the bearing assembly. Such bearings are used in large supercharging units with supports located at the ends of the rotor. Bearings of this design have a high load-bearing capacity, but are prone to self-oscillations in the hydrodynamic layer.

In turbochargers with cantilever wheels, the load on the bearings is small due to the relatively large diameter of the shaft, so they mainly use oscillating and floating bearings.

The oscillating bearing bushing is installed in the bearing assembly housing with a small gap and is secured against rotation and axial movement by a radial pin (Fig. 4.64 *c*, *d*), which often also serves as an oil supply fitting, a key whose axis is perpendicular to the bearing axis, or an outer collar clamped between the housing and the cover.

The floating bearing bushings rotate freely in the housing, their rotation speed is 25...30% of the rotor speed. Axial fixation is carried out by locking rings. A solid floating bushing is also used, the ends of which form two bearings.

The use of floating bushings reduces the relative sliding speed and increases the damping properties of the bearings due to the formation of two annular layers of oil. Therefore, swinging and floating bearings are much less prone to self-oscillations.

The disadvantage of plain bearings is increased friction losses, the value of which can be two to three times greater than that of rolling bearings. In small turbochargers these losses can amount to 10...15% of all losses. With an increase in the size of the units, the share of losses is reduced to 4...6%. As a result, in order to effectively cool the friction pairs, the oil flow through them has to be increased, which leads to the need to supply oil to the bearing under increased pressure, and the oil

For preliminary review

itself must be cooled. Otherwise, due to overheating of the bearing located on the side of the hot turbine wheel, the oil in it may overheat and coke.

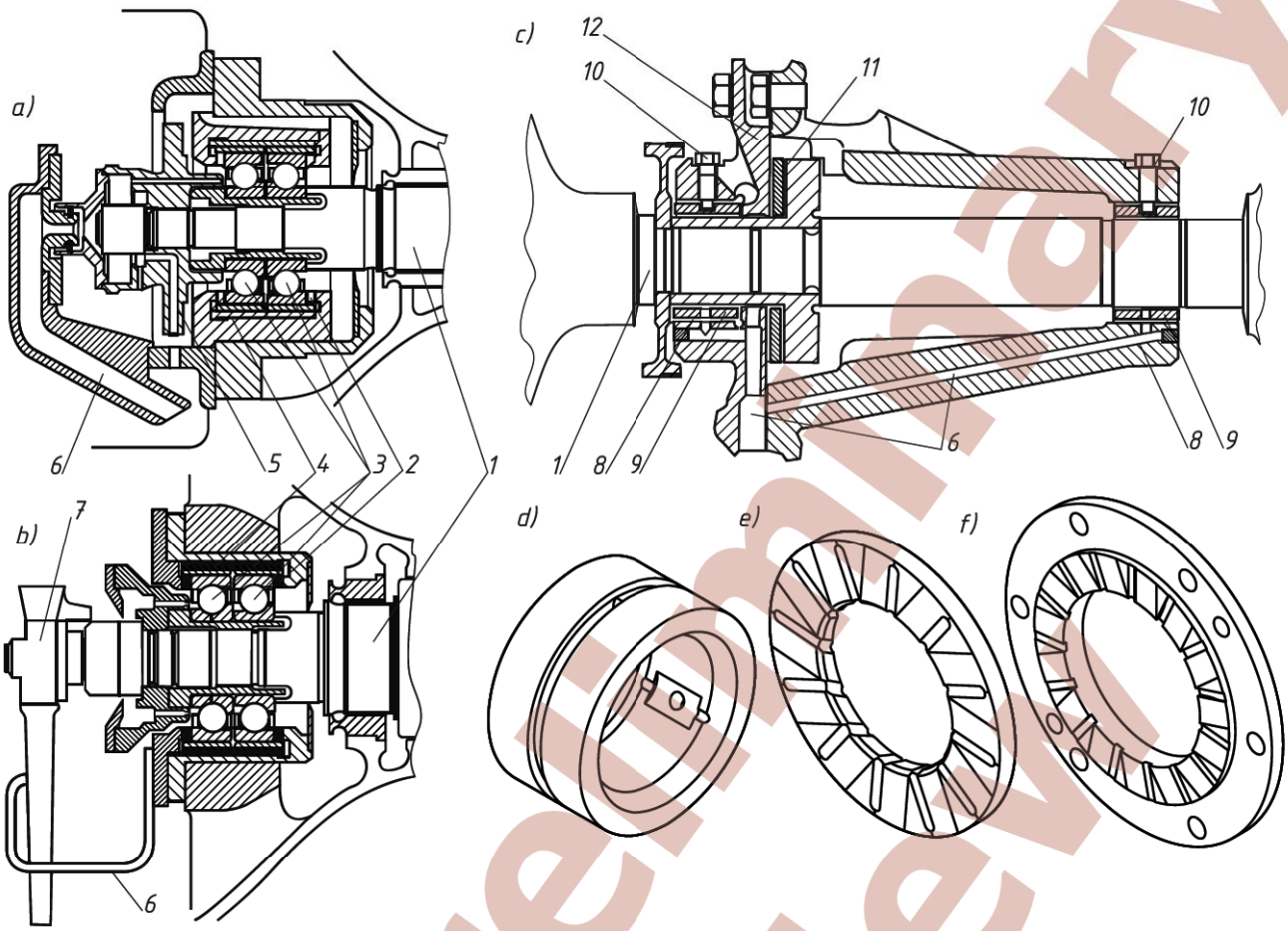


Figure 4.64 – Bearings used in turbochargers *a* – bearing assembly with rolling bearings of a VTR type turbocharger from ABB with oil supply to the bearings using a disk; *b* – the same, but with oil supply using a mounted oil pump; *c* – bearing unit with swinging radial and floating thrust bearings for a TPL type turbocharger from ABB; *d* – radial; *e* – persistent; *f* – thrust foot of the TCA series turbocharger from MAN: 1 – rotor shaft; 2 – axial elastic supports; 3 – rolling bearings; 4 – radial elastic supports; 5 – oil supply disk; 6 – channels for supplying oil under pressure; 7 – oil pump; 8 – gap between the housing and the swing bearing; 9 – swinging plain bearing; 10 – threaded pin for fixing the position of the bearing sleeve; 11 – freely rotating thrust bearing disk; 12 – bearing unit cover; 13 – suction tube (adapted from [39, 49])

When starting turbochargers after a long stay or after a sudden engine stop under load, it may be necessary to pre-bleed the bearings with oil. On large turbochargers, an autonomous circulating lubrication system is used for this, which includes an oil storage tank, a refrigerator for cooling it, a piping system, as well as oil pumps driven from the turbocharger shaft and an independent drive (usually electric).

Small supercharging units are included in the engine lubrication system, which does not in any way affect their reliability when additional oil purification is organized.

To compensate for axial forces arising as a result of the influence of gas flows on the wheels of the turbine and compressor, in addition to radial ones, thrust bearings are installed, the losses in which constitute a significant proportion of the total friction losses in the bearings of the turbocharger.

Modern turbochargers use thrust bearings with several fixed pads. Channels for lubricant supply are milled between the pads. From the channels in the direction of rotation of the thrust heel, bevels are made on the pads at an angle of 15...30', facilitating the formation of an oil wedge (Fig. 4.64 *e*). In a number of designs, the thrust bearing is made as one piece with the support bearing. To supply

For preliminary review

oil to the thrust surface, oil supply channels are milled on the inside of the bearing. Oil supply bevels can also be made on the thrust heel (Fig. 4.64 *f*). To compensate for distortions and equalize loads on the thrust surface, the latter can be installed elastically through a package of thin gaskets, the gaps between which are filled with oil.

4.7.4.7 Design features of stator elements.

Turbocharger stator consists of three main units – the compressor housing, the gas outlet housing, the turbine housing, which are connected to each other by flanges and centered by landing collars (Fig. 4.65).

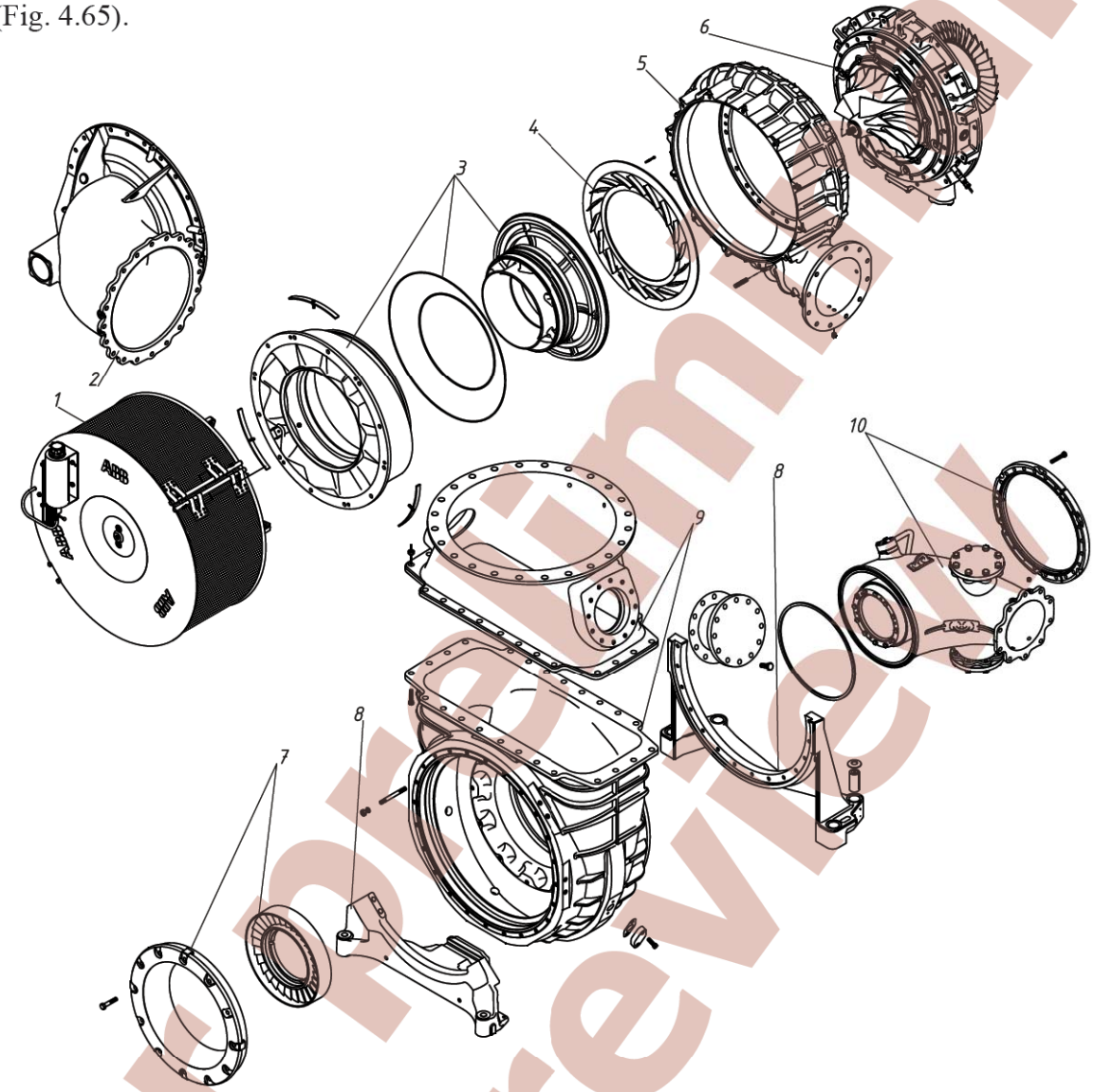


Figure 4.65 – Housing parts of a turbocharger type TPL76-C from ABB: 1 – air supply pipe; 2 – air filter; 3 – details of the sealing disc; 4 – compressor diffuser; 5 – air exhaust spiral chamber (volute); 6 – middle housing with bearings and rotor; 7 – details of the turbine inlet guide vane; 8 – turbocharger mounting supports; 9 – gas turbine housing with gas outlet pipe; 10 – details of the gas supply channel (adapted from [44])

The assembly of housings can be carried out in different relative positions of individual elements, which expands the layout possibilities of using the units. The support brackets located on the turbine housing can also be rotated relative to the turbine housing, which further facilitates the installation of the turbocharger on the engine.

Inlet and outlet pipes are connected to the housing parts.

The design of the housing parts has a decisive influence on the performance of the turbocharger. The air entry into the compressor should be smooth, without sharp turns. The supply pipe and the muffler filter must not distort the air velocity fields at the inlet.

For preliminary review

The turbine housing is cast from cast iron, and depending on the method of diesel supercharging, it can have from one to four gas intake holes.

The compressor housing is cast either from cast iron or aluminium alloy. A sealing disk and a diffuser are installed on the inside of the compressor housing, which together with the air exhaust spiral chamber form the compressor pressure line.

The air outlet chamber may have one or two connecting pipes.

A heat-insulating partition is installed between the turbine and the compressor, and the rotor shaft is covered with a heat shield. The partition and screen are made in the form of a detachable casing, which is mounted in the housing.

If the design of the turbocharger involves cooling, cavities are located in the turbine and gas exhaust housings for the circulation of cooling water.

The bearing housings have channels through which oil is supplied under pressure from the lubrication system of a turbocharger or diesel engine through pressure fittings. The cavities of the housings in which the bearings are located are separated from the cavities of the turbine and compressor by combined seals (elastic rings and labyrinths) designed to reduce leakage of charge air and exhaust gases through the gaps between the rotor and housing parts. Through drillings in the housing, air with excess pressure is supplied to the seals on the compressor side, which prevents the occurrence of vacuum in the bearing cavity and oil carryover into the compressor. Through drillings in the middle housing and the turbine housing, air also enters the cavity between the turbine wheel and the labyrinths, which helps cool the rotor and prevents hot gases from breaking through into the bearing cavity.

For turbochargers made according to a cantilever less design, each bearing unit is placed in a separate cavity, which simultaneously serves as an oil sump. The crankcases are closed at the ends with covers that have plugs for filling oil and an inspection port for monitoring its level.

4.8 Characteristics of compressors of turbocharger supercharging units

The characteristics of centrifugal radial-axial compressors are the relationship of their main operational parameters, presented in graphical or analytical form. In the most general case, the characteristics represent dependencies of the form:

$$\pi_к, \eta = f(G,\ n)\ .$$

In Fig. 4.66 shows what the characteristic of a modern marine centrifugal radial-axial compressor looks like in dimensionless parameters, which relates to each other: the degree of pressure increase by the compressor π_k (the ratio of the air pressure after the compressor to the air pressure before the compressor – $\pi_k = p_k / p_0$), adiabatic efficiency of the compressor – η_k, rotation speed of the compressor wheel (dimensionless parameter n/n_0) and volumetric air flow through the compressor – Q.

The adiabatic efficiency of the compressor – η_k takes into account compression losses associated with heating the air as a result of friction on the surface of the compressor flow part, the formation of vortices and hydraulic losses caused by the flow of air in the gaps, as a result of which the air is compressed polytropically and not adiabatically.

In Fig. 4.66 curves $n/n_0 = const$ represent the dependences $\pi_k = f(Q)$ for various constant rotor speeds, and isolines $\eta_k = const$ make it possible to determine the values of the adiabatic efficiency of the compressor for any values of π_k and volumetric air flow Q at the current rotor speed. In addition, the characteristic shows a line defining the surge limit, to the left of which the compressor cannot operate. The dimensionless characteristics parameters include: adiabatic compressor efficiency – η_k; relative rotation speed – n/n_0, degree of pressure increase – π_k. At the same time, volumetric air flow – Q, is a dimensional parameter. With different methods of presenting characteristics, the relationship between dimensional and dimensionless parameters may be different.

The compressor characteristics (Fig. 4.66) are constructed based on the results of their tests, bringing the ambient air parameters to normal (usually $p_0 = 105$ Pa, temperature $T_0 = 300$ K, relative

For preliminary review

humidity $\varphi_0 = 60\%$).

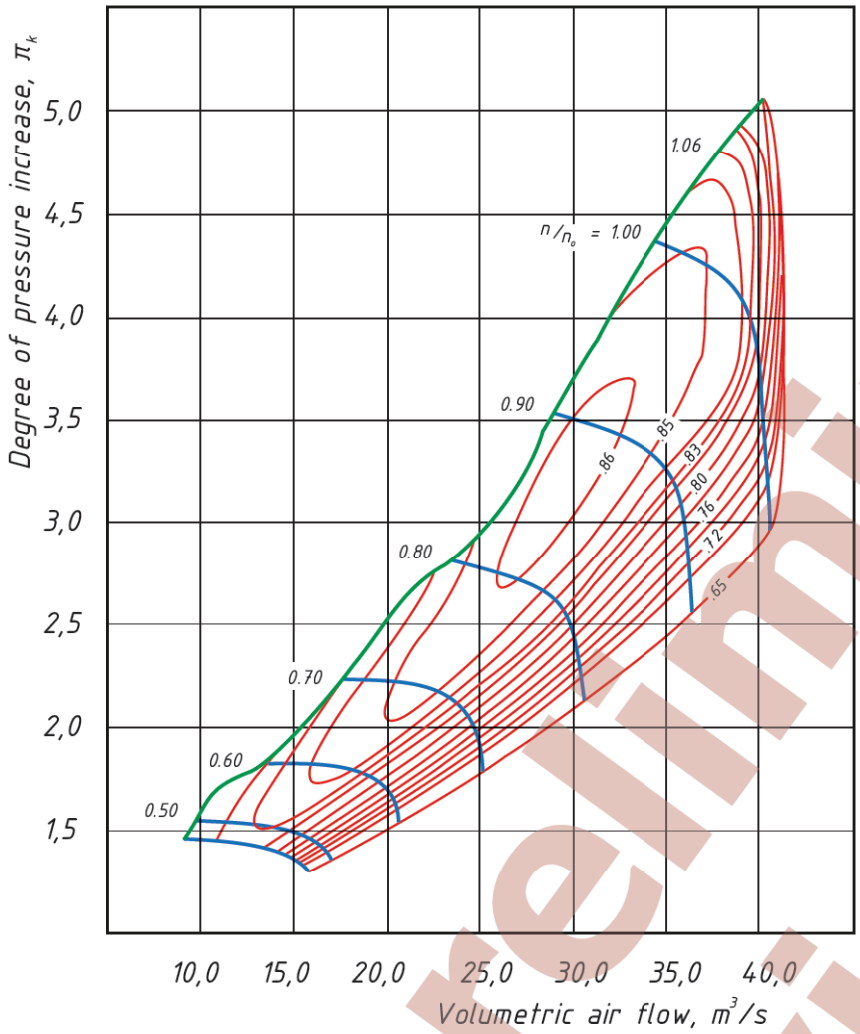


Figure 4.66 – Characteristics of the compressor of the TPL 85-B turbocharger unit from ABB in dimensionless parameters: n – rotation speed of the compressor wheel; n_0 – rated rotation speed of the compressor wheel, n/n_0 – relative rotation speed (blue lines); η_k – adiabatic efficiency of the compressor (red lines); surge limit (green line) (adapted from [50])

At the first stage of constructing the characteristics, two separate dependencies are obtained – the first connects the values of the adiabatic efficiency (η_k) with the air flow (Q) and is called the energy part of the characteristic (Fig. 4.67), and the second connects the values of the degree of pressure increase – π_k from the air flow (Q) and is called the flow-pressure part of the characteristic (Fig. 4.68).

As can be seen in Fig. 4.67, curves $\eta_k = f(G)$ for $n = const$ on the energy part, the compressor characteristics have one maximum, which corresponds to the regime in which the flow around the inlet edges of the compressor blades occurs at an optimal angle with minimal impact losses (Fig. 4.69 *b*).

Deviation from this regime, both in the direction of decreasing and increasing air flow, leads to an increase in these losses and a decrease in adiabatic efficiency. The latter is explained by the deviation of the air flow entry angle from the optimal one due to an increase or decrease in the absolute speed c_1 at the entrance to the compressor wheel (Fig. 4.69 *a*, *c*).

The right branch of the curve decreases with increasing losses, which are approximately proportional to the square of the flow rate. On the left, the branch of the curve is limited by the so-called surge boundary (point *k*), to the left of which air supply by the compressor is impossible.

Surge is a malfunction in the operation of the compressor, which leads to pressure pulsations at its outlet and shock loads on the blades as a result of periodically occurring reverse flow of gas along the flow part.

We will consider the causes and mechanism of surge occurrence below.

For preliminary review

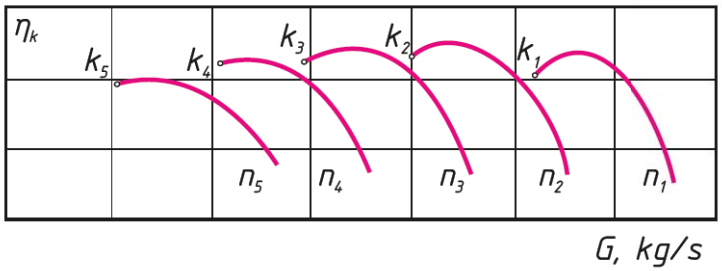


Figure 4.67 – Energy part of the characteristic. Values of adiabatic efficiency (η_k) as a function of air flow (Q) at $n = const$

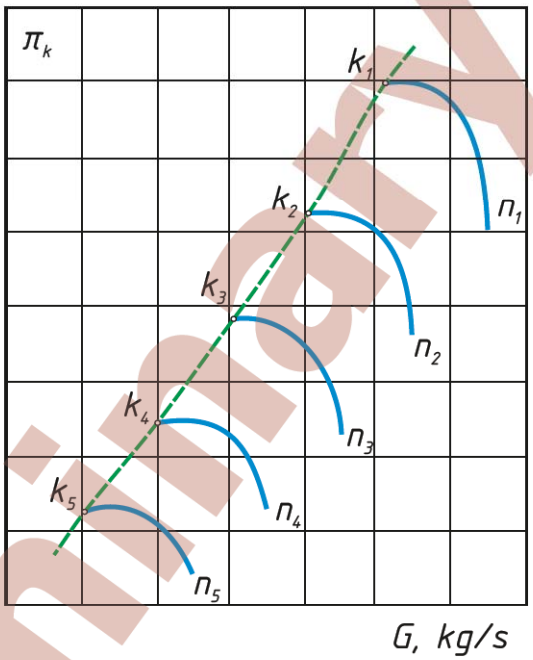


Figure 4.68 – Flow-pressure part of the characteristic. Values of the degree of pressure increase – π_k as a function of air flow (Q) at n = const

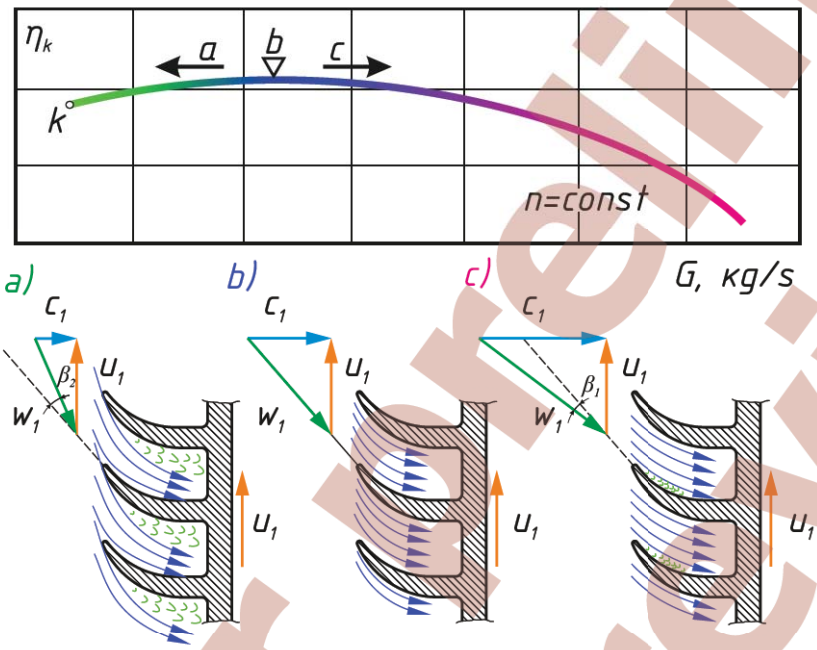


Figure 4.69 – Adiabatic efficiency of a compressor as a function of flow rate at a constant speed (top) and three cases of gas entering the channels of the compressor impeller (bottom): *a* – flow rate is less than the calculated one; *b* – design mode; *c* – flow rate is greater than calculated: *n* – isotach; *k* – point of surge occurrence (flow stall); *G* – mass air flow; *c* – absolute flow velocity; *U*, *W* – peripheral and relative speeds

The transition from the energy (Fig. 4.67) and flow-pressure parts (Fig. 4.68) to the combined characteristic (Fig. 4.66) is carried out according to the scheme presented in Fig. 4.70.

For any value of the adiabatic efficiency of the compressor, one or two values of air flow can be found by drawing a straight line (dashed line) parallel to the flow axis. In Fig. 4.70 *a*, it is clear that this straight line intersects the curves of changes in η_k for various rotation frequencies (n_5, n_4, n_3, n_2, n_1) at points $S_{5'}$, $S_{5''}$, $S_{4'}$, $S_{3'}$, $S_{2'}$, $S_{1'}$. Vertical projections of these points onto the curves of changes in π_k, for the same values of rotation frequencies, give points $S'_{5'}$, $S'_{5''}$, $S'_{4'}$, $S'_{3'}$, $S'_{2'}$, $S'_{1'}$, connecting which with a smooth line we obtain the boundary of the specified value of adiabatic efficiency on the combined characteristic. Similar curves are plotted for other efficiency values (Fig. 4.70 *b*), resulting in a combined compressor characteristic.

By connecting points k_1-k_5 with a smooth curve, we obtain a surge line (in Fig. 4.70 highlighted in green) to the left of which the compressor cannot operate.

The cause of surge is the disruption of the gas flow from the blades of the impeller and blade dif-

For preliminary review

fuser at gas flow rates that differ significantly from the calculated values. It should be noted that flow stall from the compressor blades occurs both with a decrease and with an increase in gas flow compared to its calculated value, however, the stall mechanisms and their impact on the operation of the compressor are significantly different. In Fig. 4.69 shows three cases of flow around the leading edges of the impeller blades. If in the case shown in Fig. 4.69 *c*, the resulting vortex zone is blocked by the oncoming flow in the near-wall region of the blades, then in the case shown in Fig. 4.69 *a*, the vortex zone has the opportunity to develop along the flow part of the compressor wheel, since the walls of the blades tend to expand the vortex zone. As a result, vortex zones with reduced pressure can form along the entire flow path of the compressor wheel, through which a counterflow of air periodically occurs from the air receiver, where it is under increased pressure, in the direction of the inlet pipe.

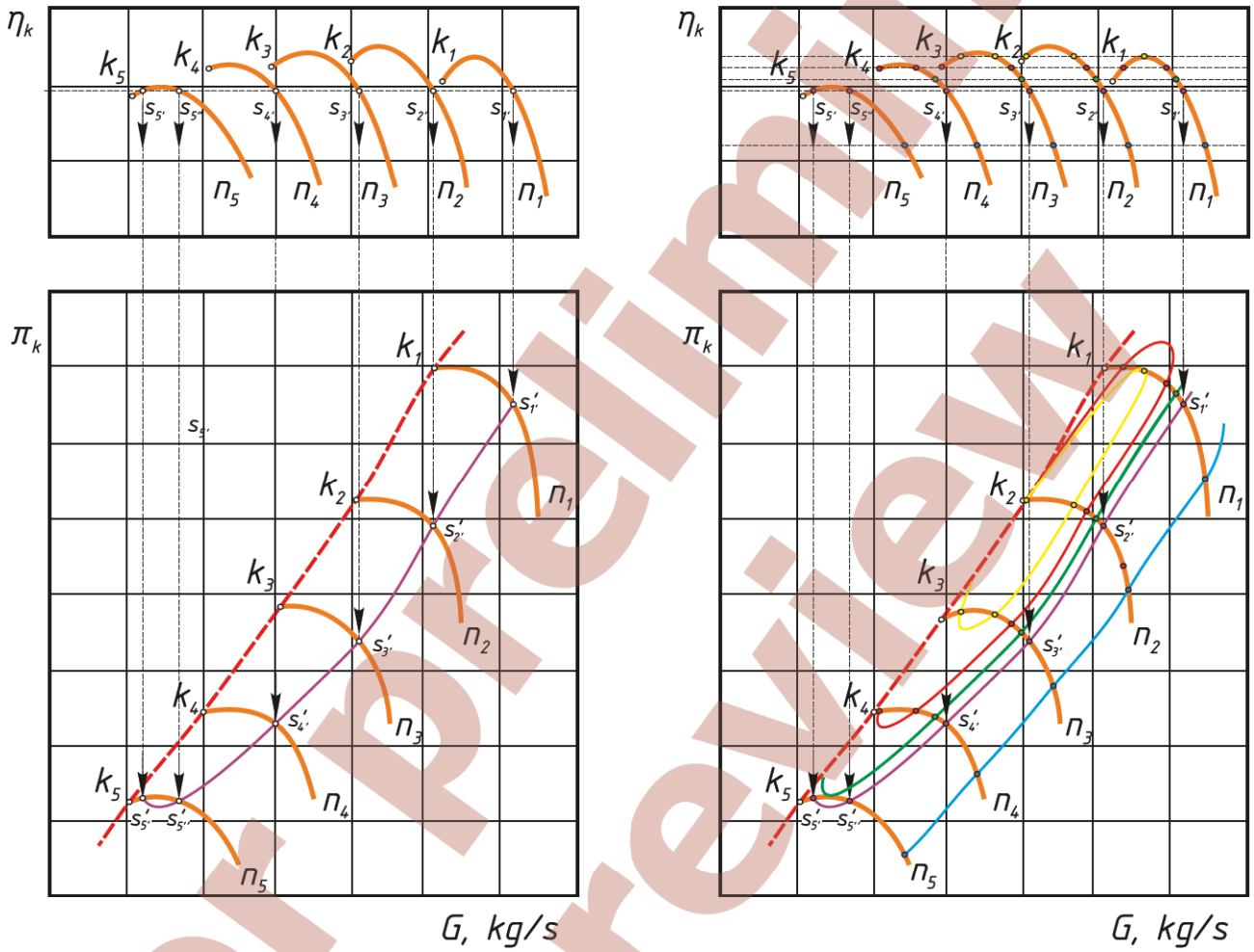


Figure 4.70 – Construction of a combined compressor characteristic by transferring adiabatic efficiency isolines to the flow-pressure part of the characteristic

When installing a vaned diffuser in a compressor, surge may occur on its blades. The mechanism of occurrence is shown in Fig. 4.71.

There are three possible cases of flow around the diffuser blades. When the speed of gas exit from the wheel corresponds to the calculated one (Fig. 4.71 *a*), the air flow expands evenly along the entire inter-blade space. With an increase in flow rate and, therefore, speed c_2, vortex zones begin to form along the convex side of the blades (Fig. 4.71 *b*), which are blocked by the oncoming flow. At low gas flow rates (reduction in speed c_2), vortex zones form at the concave walls of the blades (Fig. 4.71 *c*). The gas flow, moving along a flat trajectory, is rejected from the concave

For preliminary review

walls, which contributes to the development of vortex zones through which gas periodically flows in the direction opposite to its movement.

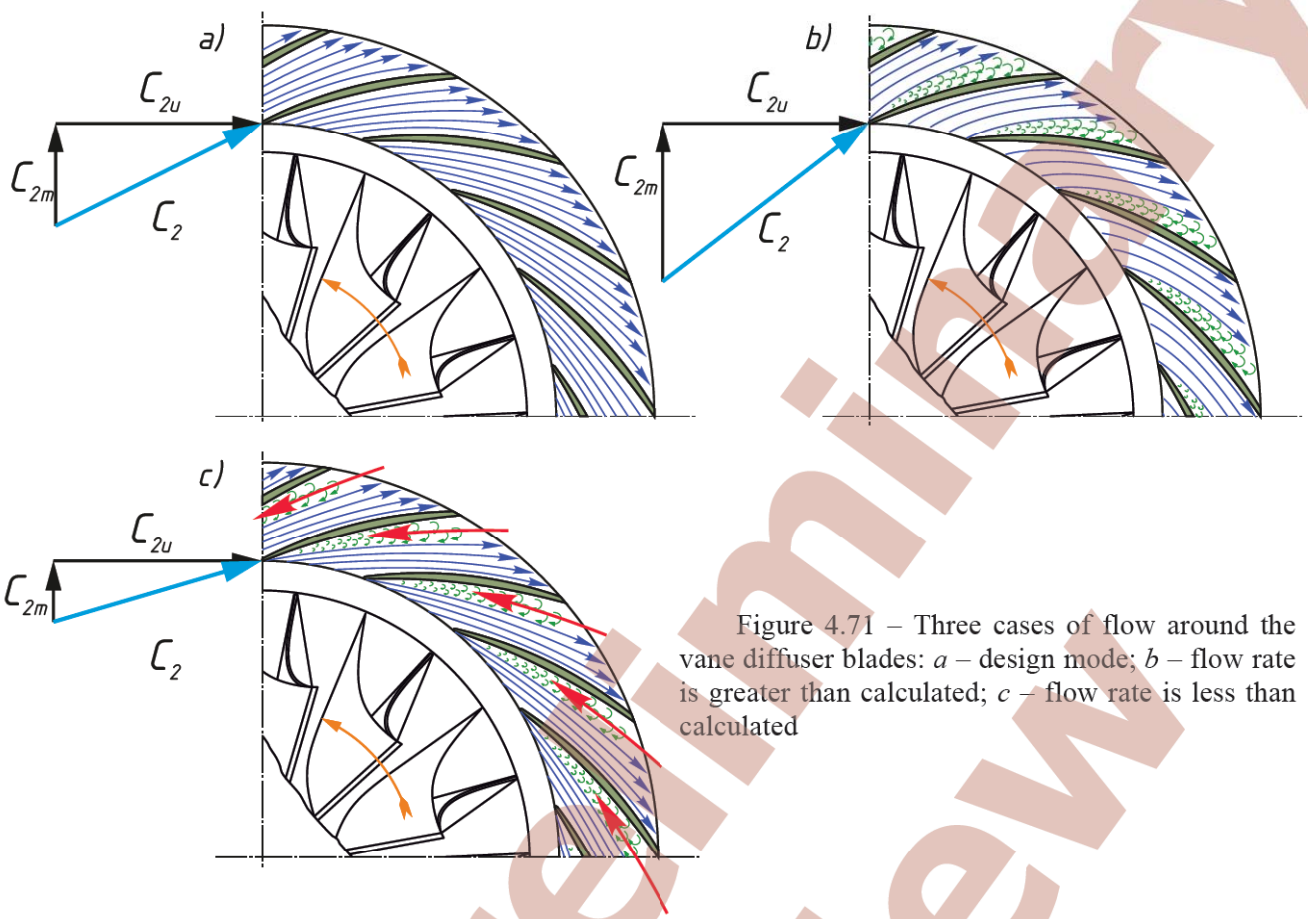


Figure 4.71 – Three cases of flow around the vane diffuser blades: *a* – design mode; *b* – flow rate is greater than calculated; *c* – flow rate is less than calculated

The absence of blades in a bladeless diffuser makes the compressor less prone to surge. In this regard, the surge line of such a compressor is shifted to the left than that of a similar compressor with a blade diffuser. Thus, the presence or absence of a vaned diffuser in a compressor affects the shape of its characteristic. All other things being equal, the presence of a bladed diffuser makes it possible to increase the efficiency of the compressor at the design mode and a narrow zone of adjacent modes. In other areas, remote from the design air flow rate, a drop in efficiency will be observed in relation to the bladeless version.

During surge, pulsations of the gas flow cause vibrations of the blades, which can lead to their damage. For this reason, operation of the compressor in surge mode is unacceptable due to its possible failure. To exit the surge mode, it is necessary to increase the gas flow through the compressor. To do this, an anti-surge valve is installed immediately behind the compressor, which bypasses part of the charge air into the exhaust receiver.

It should be noted that the line of stable compressor surge operation is at the same time a limiting line for the range of available engine operating modes under operating conditions. Thus, the characteristics of compressors are necessary to determine the parameters for the joint operation of the compressor unit and the engine. With their help, you can determine the range of limit values of the main operating parameters that a given compressor can provide. Within the boundaries of this area, there is an infinite number of combinations of these parameters, in which material and energy balances are established between the compressor and the engine. With this balance, the air flow through the engine must be equal to the air supplied by the compressor, and the degree of pressure increase by the compressor must be equal to the increase in pressure of the charge air in front of the engine. To determine the conditions for the existence of balance during joint operation of the engine and compressor, the concept of the so-called gas-dynamic characteristics of the engine is intro-

For preliminary review

duced. This relationship between the air flow through the engine and the degree of pressure increase in front of it, which is necessary to ensure a given flow. If we take into account that the degree of pressure increase on the engine (π_{eng}) is the ratio of the air pressure in front of the engine (p_{eng}) to the ambient air pressure (p_0): $\pi_{eng} = p_{eng}/p_0$, and also, then what, the pressure in front of the engine is equal to the pressure after the compressor: $p_{eng} = p_k$, it follows that $\pi_{eng} = \pi_k$.

A general view of the gas-dynamic characteristics of the engine for a given crankshaft speed and a fragment of the compressor characteristics for a given rotor speed are presented in Fig. 4.69.

For the case considered, the condition of energy and material balances will be met by the point of intersection of these characteristics, plotted in the same coordinates (Fig. 4.72). This point is called the point where the engine and compressor work together, or the operating point.

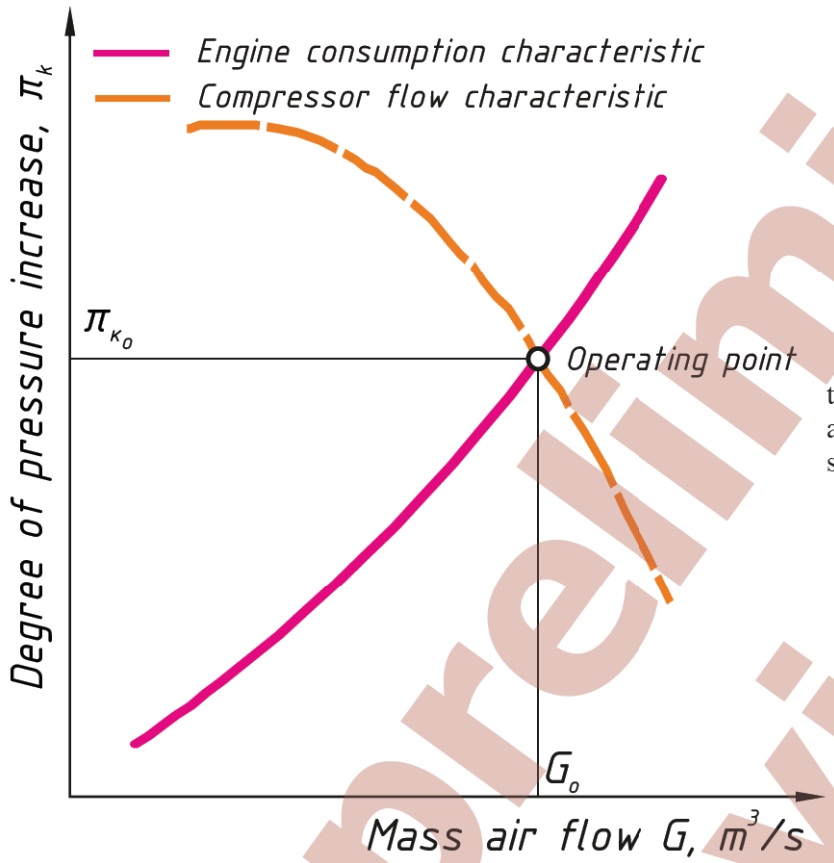


Figure 4.72 – Determination of the operating point when the engine and compressor operate together at steady state

The presented approach to determining the operating point is the most general and does not take into account many factors that must be taken into account when determining the parameters of the joint operation of the turbocharger and engine.

Such factors include changes in the rotation speed of the compressor rotor, the ability to provide the required torque on the turbocharger rotor taking into account the characteristics of the gas turbine, engine operating process parameters, characteristics of its loading, etc.

In Fig. 4.73 shows the combination of consumption characteristics of a four-stroke marine engine and a turbocharger when the engine operates according to different characteristics, absolute external, load and screw. In this characteristic, the wheel rotation speed is replaced by the value of the peripheral speed U_2.

With an increase in air flow through the compressor and, accordingly, a transition to larger wheel diameters, the dimensions of all other elements of the flow path increase. This makes it possible to better profile and process the surfaces of elements that directly interact with the gas flow, which contributes to better gas-dynamic processes and, accordingly, an increase in the efficiency of the compressor as a whole. This is clearly seen from Fig. 4.66 and 4.73. The compressor, the characteristics of which are shown in Fig. 4.66, operates in the area of higher flow rates and is larger than the compressor whose characteristics are shown in Fig. 4.73, accordingly it has greater efficiency.

For preliminary review

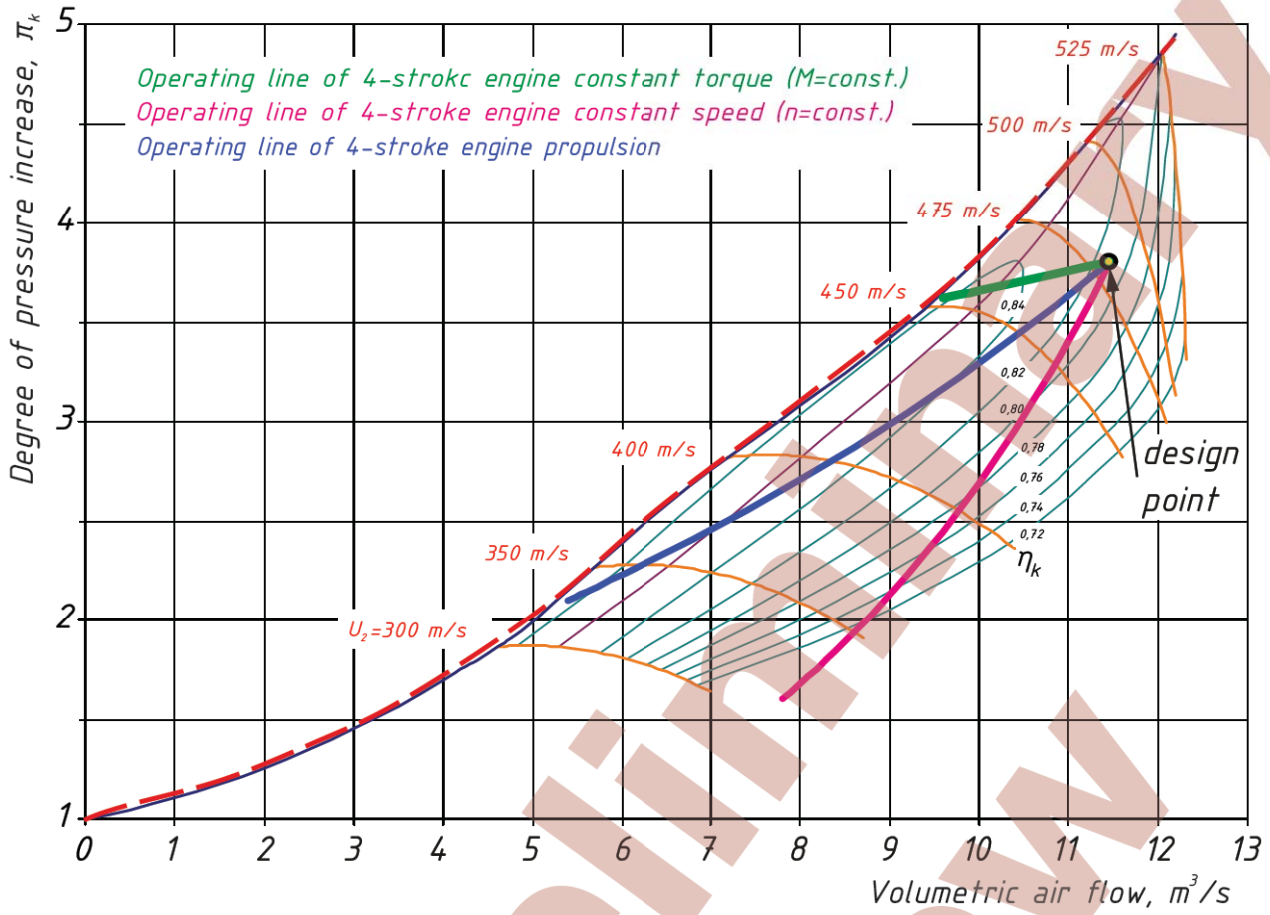


Figure 4.73 – Combined characteristic of the compressor of the NA 40/S turbocharger unit combined with the consumption characteristics of a four-stroke engine under different loading methods (adapted from [43])

4.9 Adjustment of charging units

As follows from the previous section, an important aspect of the effective use of gas turbine supercharging is the coordination of the characteristics of the supercharging units and the piston part of the engine. It should be noted that such coordination is a rather difficult task even in steady-state operating modes and is especially difficult in transient ones. As a rule, harmonization of characteristics in any mode is achieved by reducing the efficiency of the combined engine in other modes. If the engine operates over a wide range of loads, it often becomes necessary to regulate the charging units.

Many modern engines, and especially four-stroke ones, at modes close to maximum power, have a significant reserve of exhaust gas energy. To match the power of the turbine with the power consumed by the compressor, they often resort to bypassing part of the exhaust gases past the turbine directly into the exhaust system (the so-called bypass) through a special bypass channel. When engine power decreases, the loss of turbine power is compensated by increasing the supply of gases to it by reducing their bypass. For this purpose, the bypass channel is equipped with a special adjustable valve.

In a number of designs, the turbine power is not regulated, and the excess air that the compressor pumps is bypassed bypassing the engine into the exhaust manifold.

Regulation by bypassing exhaust gases or air is the simplest way to match characteristics, but this increases unproductive energy losses and reduces the overall efficiency of the diesel installation.

As already noted, to match the characteristics of turbochargers with the characteristics of the

For preliminary review

engine over a wide range of loads, inlet guide vane devices with varying geometry are used. Such regulation can be more effective than bypassing, but the design of turbochargers is significantly more complicated. The geometry can be changed either stepwise or smoothly by rotating the guide vane array blades.

An example of a radial turbine with a stepwise change in the settings of the turbine guide vane is shown in Fig. 4.74.

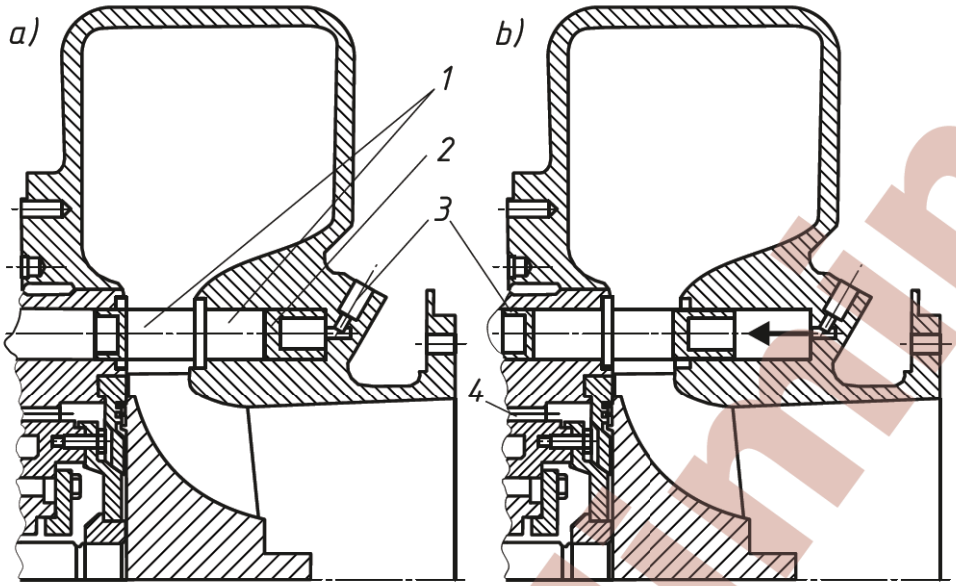


Figure 4.74 – Turbocharger with a stepwise change in the geometry of the guide vane apparatus: *a* – partial load mode; *b* – mode close to full power. 1 – sets of blades with different flow sections; 2 – pneumatic piston; 3 – control air supply channel; 4 – sealing air supply channel

When working at low loads, a guide vane apparatus with a small flow area is used, and when moving to full loads, a device with large flow sections is used. Both sets of blades are located in the turbine housing and are moved by a pneumatic piston depending on the engine load.

Mitsubishi Heavy Industries has developed its own version of stepwise regulation of turbochargers when switching to partial load modes for its MET series turbochargers. This technology is called Variable Turbine Inlet (VTI). The basic operating principle of the system is shown in Fig. 4.75 and 4.76. For stepwise regulation, the turbine of the charging unit is equipped with a two-level guide vane grid.

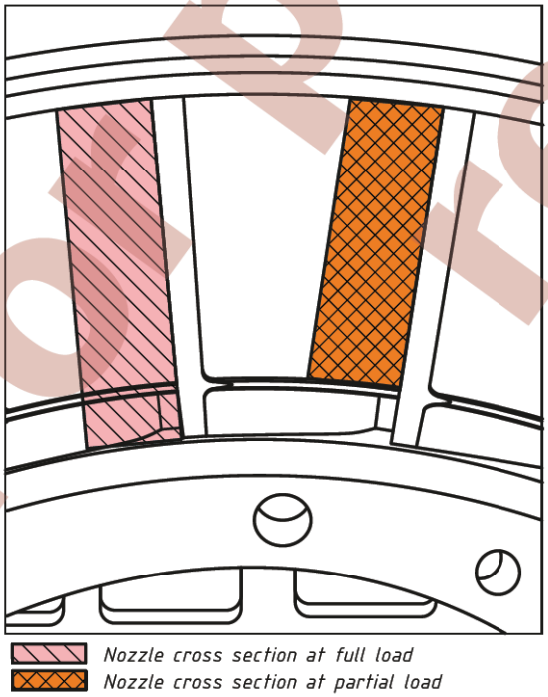


Figure 4.75 – Two-level nozzle array of MET series turbochargers from Mitsubishi Heavy Industries equipped with VTI system (adapted from [51])

For preliminary review

A partition is installed grid the levels of the grille, which divides its frontal surface into two annular areas – internal and external (Fig. 4.76).

The internal area makes up approximately 10% of the total area of the front surface of the grid.

Each area of the inlet guide vane has a separate supply of exhaust gases from the main and auxiliary chambers, connected to each other by a bypass gas channel. The gas valve equipped with the bypass channel can block the access of gases to the auxiliary chamber.

At full load modes, the gas damper opens and the engine exhaust gases enter both gas chambers, and from them through the guide vane in two parallel flows to the turbine wheel.

As the engine load decreases, the amount of gases entering the turbocharger decreases. To increase the air supply to the engine, the gas damper closes, and gases enter the turbine wheel only from the main gas chamber through the outer area of the inlet guide vane. As a result, the speed of gases at the entrance to the turbine wheel increases, the compressor speed increases and the amount of supplied air increases.

The main advantage of this solution is the simplicity of the design, the absence of moving elements of the inlet guide vane, which allows optimizing the profiles of its blades, especially in the areas where they adjoin the inner and outer rings, as well as generally increasing the reliability of the inlet guide vane.

The disadvantage is a slight decrease in the efficiency of the turbine at full load modes, which is explained by the presence of additional resistance to the gas flow in the form of a partition between the levels.

It should be noted that VTI technology can be used as an additional option when producing new turbocharger units, or for upgrading those already in operation. In this case, the guide vane grille itself and the gas distribution chamber, which is made in the form of a separate structural element attached to the turbine housing via a flange connection, must be replaced (Fig. 4.77).

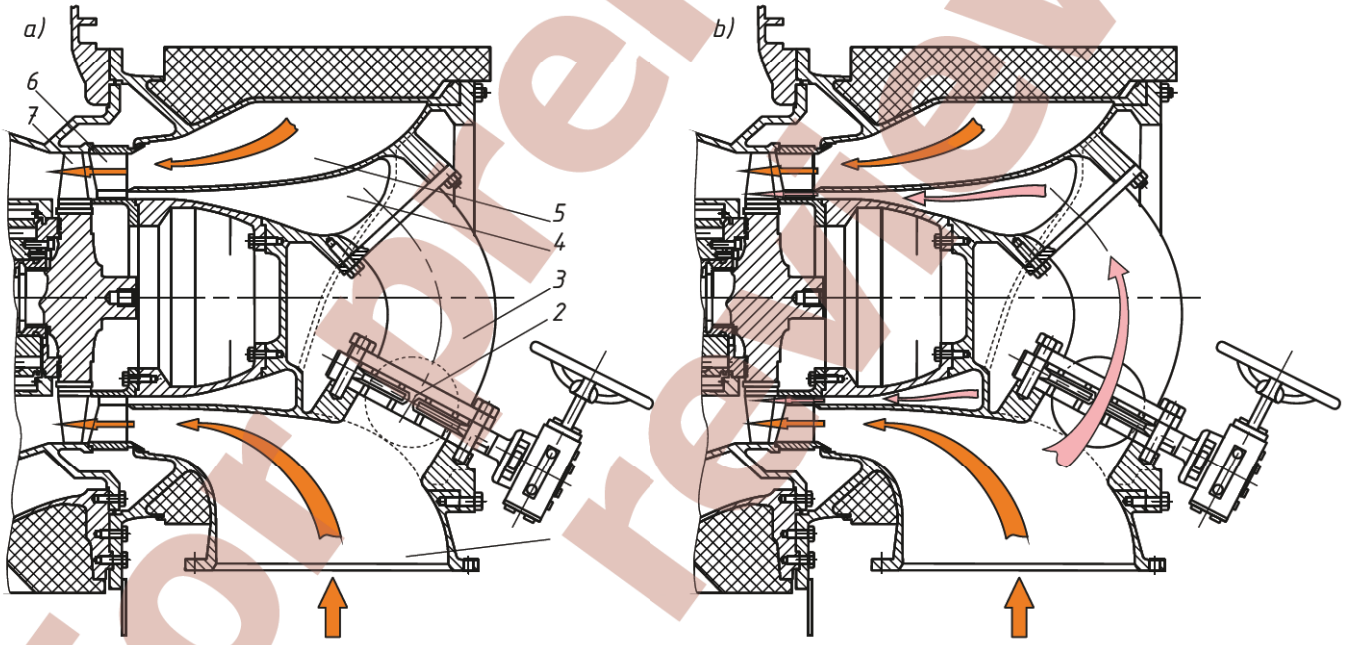


Figure 4.76 – Diagram of the organization of gas flows in the inlet guide vane of a turbine of MET series turbochargers from Mitsubishi Heavy Industries equipped with the VTI system: 1 – gas inlet pipe; 2 – gas damper; 3 – gas bypass channel; 4 – additional gas chamber; 5 – main gas chamber; 6 – inlet guide vane grille; 7 – turbine wheel (adapted from [51])

It should be noted that VTI technology can be used as an additional option when producing new turbocharger units, or for upgrading those already in operation. In this case, the inlet guide vane grille itself and the gas distribution chamber, which is made in the form of a separate structural element attached to the turbine housing via a flange connection, must be replaced (Fig. 4.77).

For preliminary review

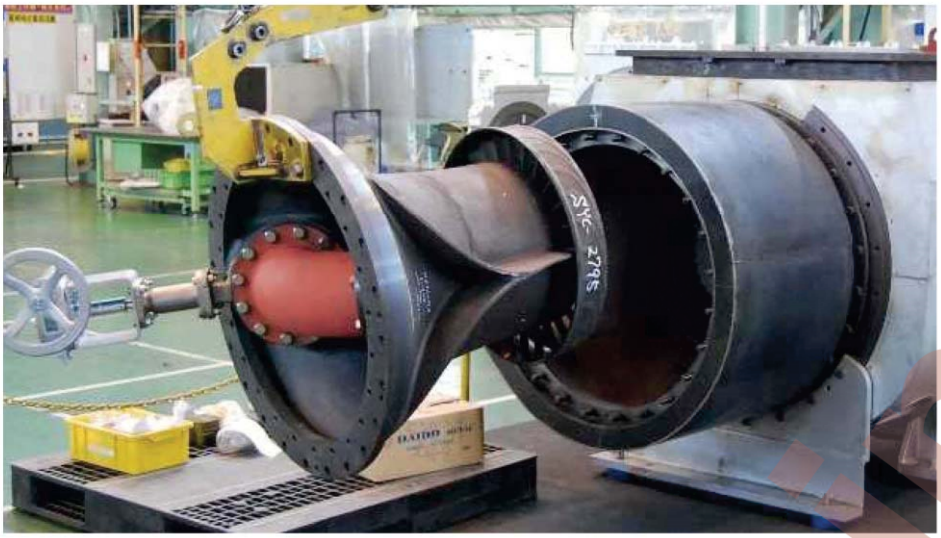


Figure 4.77 – Gas distribution chamber of the VTI system of MET series turbochargers from Mitsubishi Heavy Industries (adapted from [51])

A common disadvantage inherent in step control systems is the inability to flexibly change the characteristics of the supercharging unit, optimizing them for the engine operating mode over a wide range of load changes.

To smoothly regulate the operating mode, some manufacturers of turbochargers equip them with adjustable nozzle grilles with rotating blades, which allow you to change the speed and angle of exit of the gas flow onto the turbine impeller. This makes it possible to regulate the operating mode of the turbocharger depending on the amount of gases coming from the engine.

Different manufacturers, developing the technology of variable turbochargers, give them different names. So ABB is introducing these technologies under the name Variable turbine geometry (VTG), and MAN – Variable Turbine Area (VTA).

It is easiest to organize smooth control of the gas flow at the turbine wheel inlet in a radial-axial turbine, since the nozzle blades rotate between two fixed planes perpendicular to the axis of their rotation, so the axial clearances between the fixed surfaces and the blades remain constant. This control method is used, for example, in turbochargers of the TPS series from ABB (Fig. 4.78), as well as in turbochargers of the TCR series manufactured by MAN.

In axial turbines, the design of the inlet guide vane is more complex, but leading manufacturers of turbochargers for marine engines make such complications in order to improve the efficient and environmental performance of diesel engines.

Figure 4.78 – Adjustable inlet guide vane of a radial-axial turbine (adapted from [52])

For preliminary review

The general view and design of the inlet guide vane of MAN TCA series turbochargers with the VTA system is shown in Fig. 4.79.

To rotate the blades, drive mechanisms are used, which are mechanically driven by a control device. The drive consists of two electric motors, which drive two linear gearboxes of the screw-nut type through a cardan transmission (Fig. 4.80). On the driven element of the gearbox (nut), there is an oval hole into which the studs of the rotating ring enter with holes for the carriers of each blade.

To ensure a constant flow area, the mating of the blades with the cylindrical shank is made in the form of spheres, with the center of the spheres lying on the continuation of the axes of rotation of the blades.

A similar approach to creating adjustable grids for axial turbines of the TPL series is being developed by ABB within the framework of VTG technology.

When the load on the engine decreases, changing the angle of gas entry onto the turbine wheel blades makes it possible to increase the rotation speed of the turbocharger, thereby increasing the amount of air it supplies to the engine.

As a result, the boost pressure increases and the cleaning conditions for the working cylinders improve, especially in two-stroke engines.

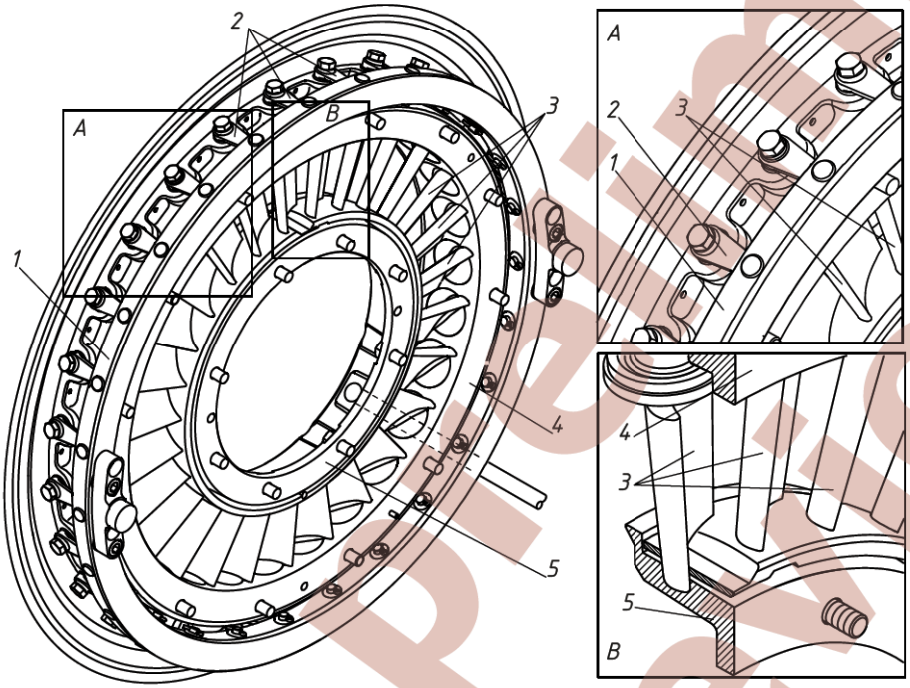


Figure 4.79 – Inlet guide vane of an axial turbine of a VTA type turbocharger unit from MAN with an adjustable inlet guide vane: 1 – annular carrier; 2 – blade rotation levers; 3 – shoulder blades; 4 – outer ring; 5 – inner ring (adapted from [53])

With a constant fuel supply, an increase in the mass of the air charge leads to improved combustion conditions and completeness, and a decrease in the maximum cycle temperature. As a result, the content $_{of}$ incomplete combustion products CH, CO, soot (C), as well as nitrogen oxides (NO_x) in the exhaust gases is reduced. Taken together, these factors lead to a reduction in engine fuel consumption (Fig. 4.81). The disadvantage of the adjustable inlet guide vane of axial turbines is the presence of a complex geometry of the transition from the blade to the shank; as a result, the resistance of a guide vane with variable geometry at nominal mode is higher than that of a guide with a constant geometry. For this reason, the efficiency of a turbocharger with a variable grille at full load is lower than that of similar units not equipped with these devices (Fig. 4.81). When switching to partial loads, the efficiency of the turbocharger increases significantly. A particularly effective system for changing the geometry of the inlet guide vane can be achieved when it is integrated into the overall engine control system with electronic control systems. A complex effect on the parameters of air supply, fuel supply and lubrication of working cylinders can significantly improve engine performance, especially in transient modes and partial load modes.

In addition, for low-speed engines, the presence of variable geometry technology allows, when switching to low-speed modes, to launch additional electric air blowers at lower loads.

For preliminary review

An important sector for the application of controlled charging technology is gas and gas-diesel engines, in which the conditions of knock-free combustion greatly depend on the ratio of the components of the gas-air mixture.

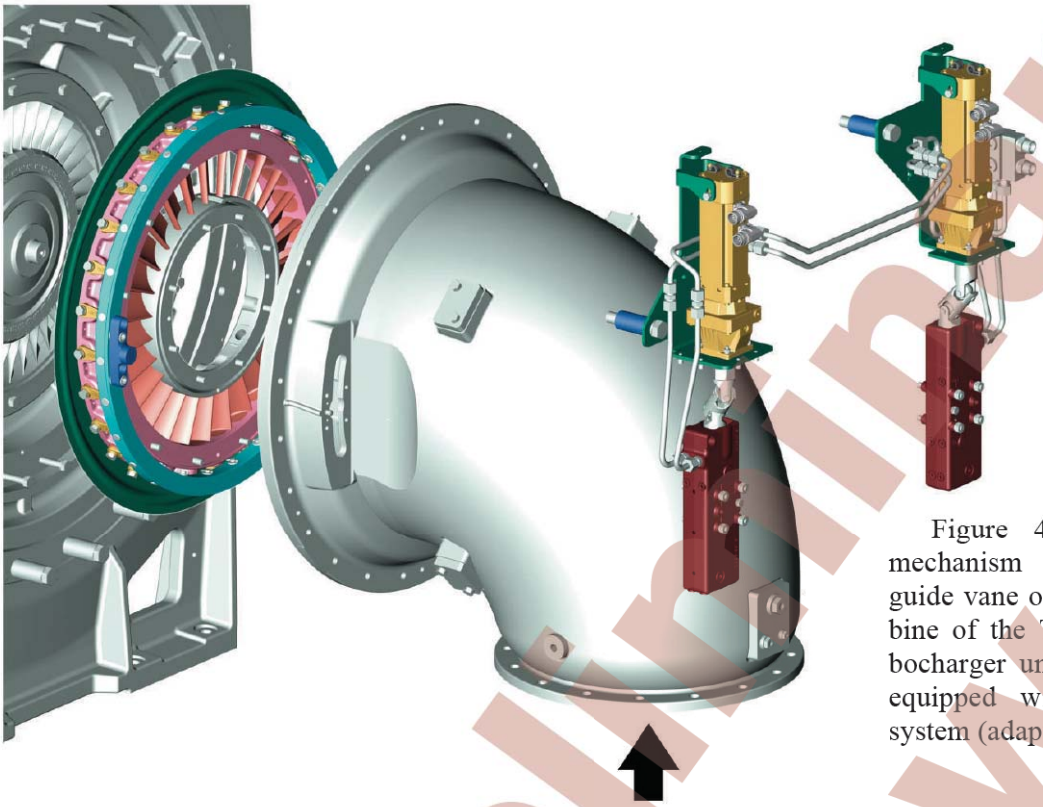


Figure 4.80 – Drive mechanism of the inlet guide vane of the axial turbine of the TCA type turbocharger unit from MAN equipped with the VTA system (adapted from [53])

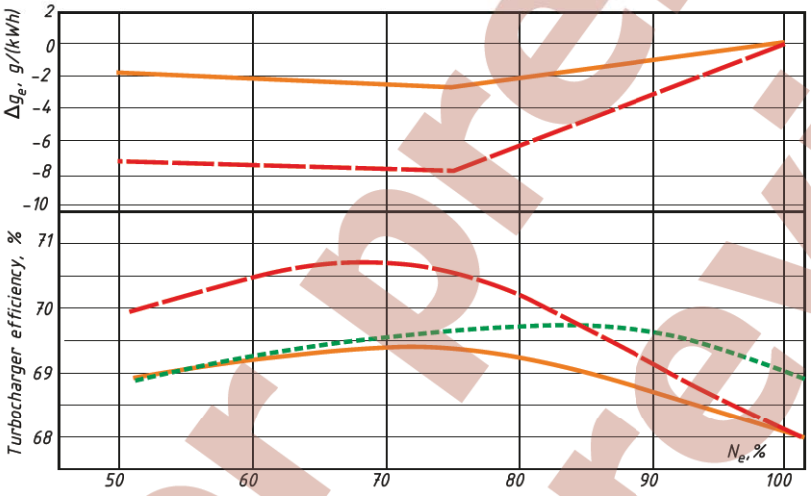


Figure 4.81 – Change in specific fuel consumption and efficiency of the TCA55-2 turbocharger unit when using VTA technology on a low-speed MAN 4T50ME-X engine. —— – characteristic with the maximum opening angle of the inlet guide vane; – – – – with the maximum closing angle of the inlet guide vane; - - - - – characteristic with a fixed inlet guide vane (without VTA) (adapted from [53])

The most difficult process is matching the characteristics of the turbocharger and the piston part of the engine during transient conditions.

The lack of a mechanical connection between the diesel crankshaft and the turbocharger rotor leads to the fact that when the load changes, the turbocharger reacts with a delay due to the inertia of both gas flows and mechanical parts. So, as the load increases, the fuel pumps increase the fuel supply to the engine, and the turbocharger increases the air supply with a lag. In this case, the optimal air-fuel ratio is disrupted, and the efficiency of fuel combustion decreases. Engine operation is accompanied by smoke.

In some designs, to reduce the acceleration time of the inflating unit, compressed air is supplied to the compressor blades from the ship's general compressed air system, using additionally installed nozzles in the sealing disk (Fig. 4.82). The air supply control system is combined with the fuel

For preliminary review

pump control system. As the load increases, the speed controller tends to move the fuel rack in the direction of increasing fuel supply. In this case, the control system limits the movement of the rack and at the same time opens the solenoid valve supplying air to the compressor blades. As the turbocharger spins up, the control system removes restrictions on the position of the fuel rack and reduces the air supply until it completely stops.

When a diesel engine suddenly drops the load, the rotor, by inertia, continues to pump air into the intake manifold, while the engine does not have time to consume the incoming air, which leads to an increase in pressure in the intake manifold. This can lead to disturbances in the conditions of air flow through the channels of the compressor wheel and cause surging.

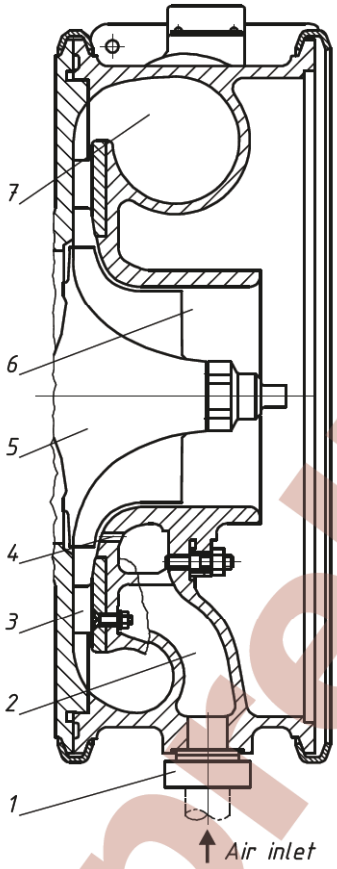


Figure 4.82 – System for tightening the compressor with compressed air of the NR 14/S turbocharger from MAN: 1 – air supply pipe; 2 – air cavity; 3 – diffuser; 4 – nozzle for supplying compressed air to the compressor wheel (adapted from [54])

More flexible control of the charging system is possible when using electric turbochargers. These devices are a unit in which a turbine, compressor and electric machine are placed on one shaft. An example of such an inflating unit from Mitsubishi is shown in Fig. 4.83. Excess turbine power, which is present in modes close to full load, is spent on generating electrical energy. This energy goes through a system of converters into the general electrical network. At partial load modes, and especially during transient modes associated with load surge, the electric machine, taking energy from the ship's general network, spins the compressor to the required speed. When the load is suddenly dropped, the electric machine switches to rotor braking mode, reducing its speed in proportion to the decrease in load. The control functions of an electric turbocharger unit can be assigned to both a separate control system and the engine control system as a whole. When using supercharging units of this type, there is no need for electric blowers.

For the first time, a hybrid electric turbocharger (Fig. 4.83 *a*) was installed on the bulk carrier «Shin Koho», equipped with a MAN 7S65ME-C two-stroke diesel engine with a power of 20090 kW. The standard turbocharger was retrofitted with a permanent magnet synchronous electric machine (Fig. 4.83 *b*), installed on the compressor side, as shown in Fig. 4.83 *a*. As a result, the turbocharger unit has become 313 mm longer and 4600 kg heavier compared to the base turbocharger. The maximum design power output was 754 kW at about 9000 min^{-1}, which provided an additional 3.75% of the main engine's power output. The generated electricity fully covered the entire needs of the vessel, which during sailing mode did not exceed 420 kW.

For preliminary review

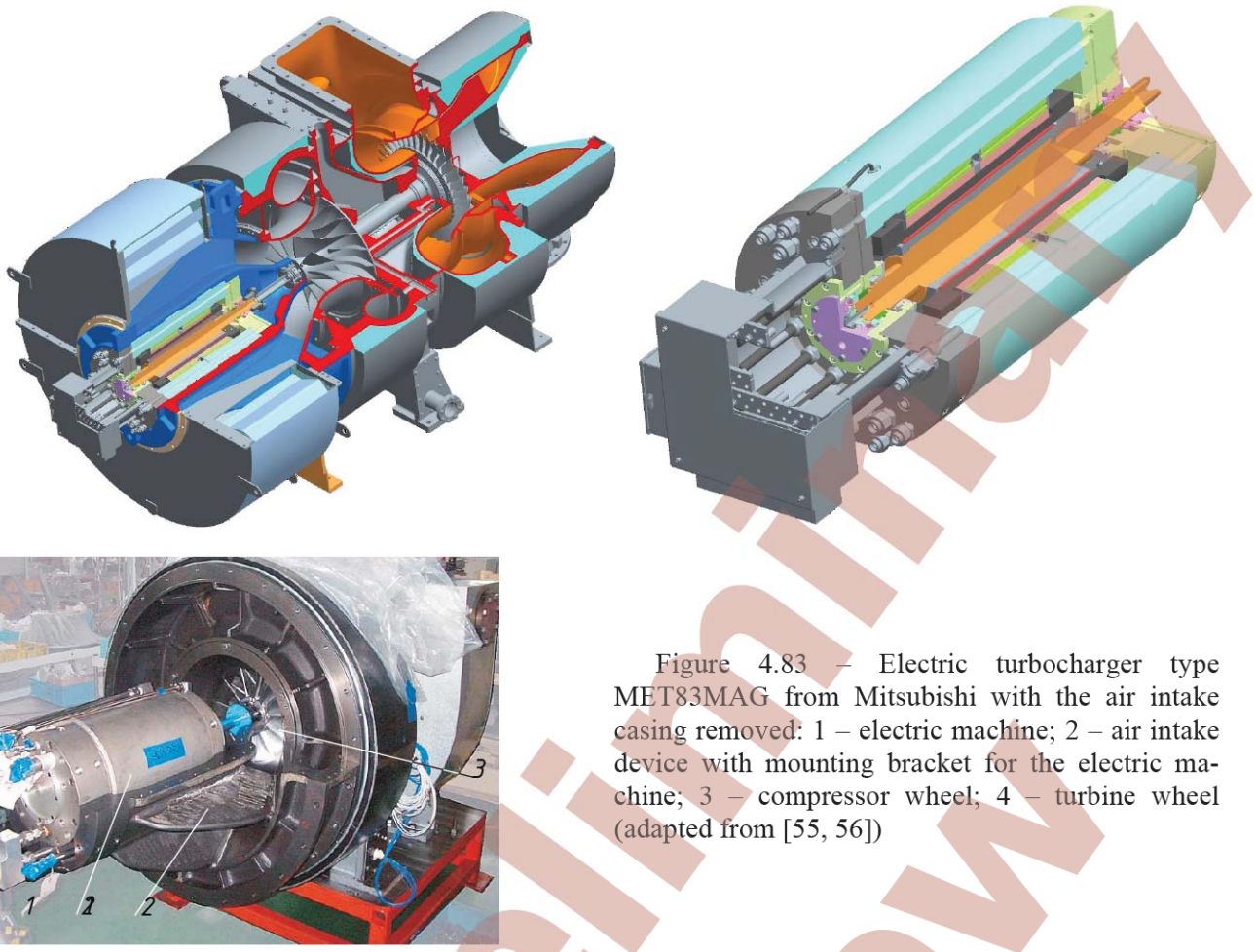


Figure 4.83 – Electric turbocharger type MET83MAG from Mitsubishi with the air intake casing removed: 1 – electric machine; 2 – air intake device with mounting bracket for the electric machine; 3 – compressor wheel; 4 – turbine wheel (adapted from [55, 56])

As already noted, low-speed engines have an excess of exhaust gas energy when operating at conditions close to nominal. This leads to the need to divert part of the exhaust gases past the turbine in order to balance the power of the turbine and compressor. Typically, the gases exhausted past the turbine are fed into a recovery boiler, where their thermal potential is used to generate steam for general ship needs. However, there are solutions in which the excess energy of the exhaust gases is used directly to generate mechanical work on the engine shaft. An example of such a system developed by Kawasaki and designated K-GET (Kawasaki-Green Eco Turbine) is shown in Fig. 4.84. The system consists of an additional gas turbine specially optimized for operation as part of the installation, a gearbox that allows the power generated by the turbine to be transmitted to the crankshaft of a piston engine, and a gas pipeline system equipped with appropriate remote-controlled valves. When the engine operates at conditions close to nominal, the shut-off valves, controlled by the electronic engine control unit, open the bypass channel through which gases from the exhaust receiver enter the power turbine. Having worked in the turbine, the gases return through another pipeline to the common gas duct of the engine after the turbocharger. The additional energy generated by the gas turbine is transferred to the engine shaft through a gearbox. Such systems are called turbocompound systems. Their use allows reducing fuel consumption by the installation by 2...3%.

A simpler solution to the problem of matching the characteristics of the compressor and the piston part of the engine in a wide range of load-speed conditions is to use a combined compressor drive. In low power modes, in which the energy of the exhaust gases is not enough to drive the compressor, the shaft of the compressor unit is driven from the engine crankshaft through a gear multiplier and a freewheel mechanism.

The gear multiplier consists of a row gear and a planetary gear. The latter, depending on the ratio of power flows, can operate in differential or planetary multiplier mode. With an increase in engine power, the power of the gas turbine increases, and the share of mechanical energy spent on driving the compressor unit decreases until the power of one gas turbine is sufficient to drive the

For preliminary review

supercharger. In this case, the mechanical drive is switched off using a roller freewheel.

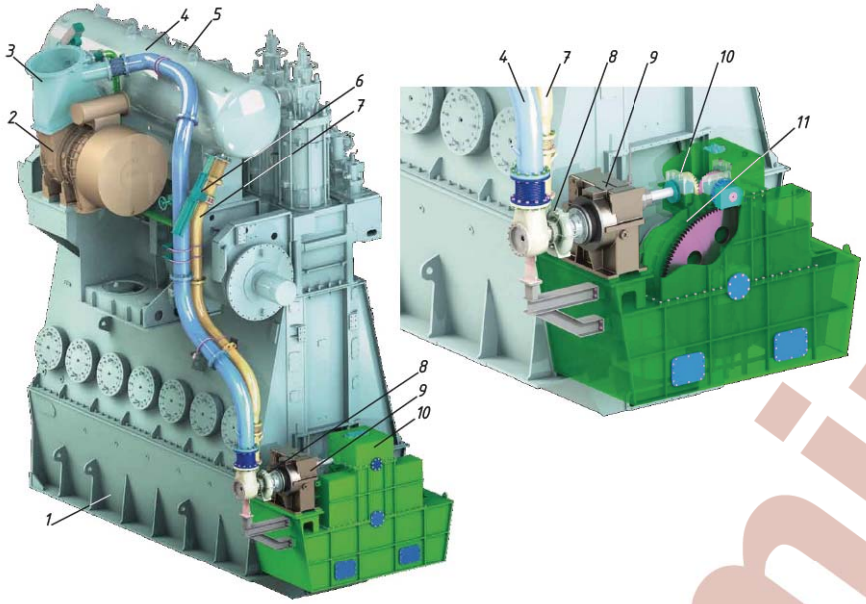


Figure 4.84 – Kawasaki K-GET turbocompound system: 1 – main engine; 2 – turbocharger; 3 – gas duct; 4 – return pipeline for exhaust gases in the recovery turbine; 5 – exhaust receiver; 6 – damper of the bypass channel; 7 – gas supply pipeline to the recovery turbine; 8 – recovery turbine; 9 – first stage gearbox; 10 – elastic coupling; 11 – second stage gearbox (adapted from [57])

This type of charging units is used in its two-stroke high-speed engines by Electro-Motive Division. A section of the inflating unit of the EMD 710 engine of this company is shown in Fig. 4.85.

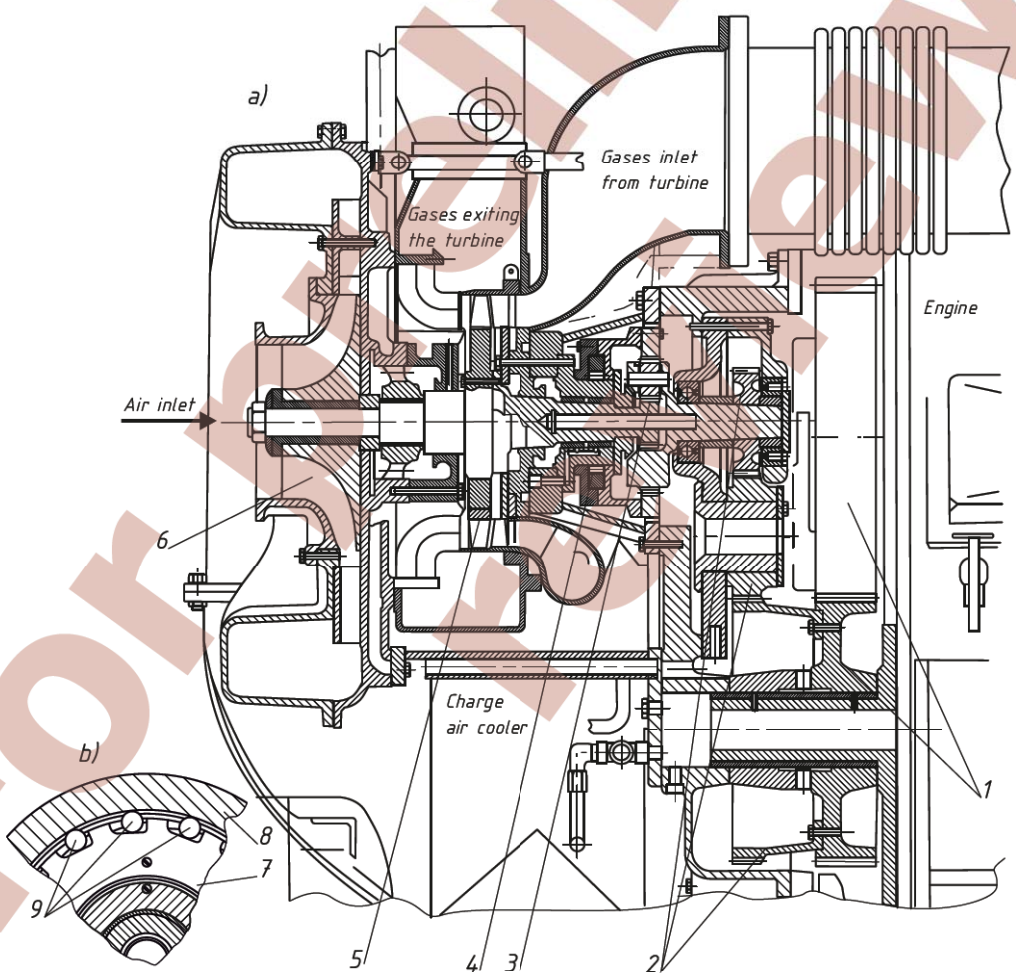


Figure 4.85 – Section of the inflating unit of the EMD 710 engine from Electro-Motive Division (*a*) and freewheel (*b*): 1 – camshaft drive gears; 2 – gear wheels of the animator gear; 3 – planetary multiplier mechanism; 4 – roller freewheel; 5 – turbine wheel; 6 – compressor wheel; 7 – coupling rotor; 8 – coupling stator; 9 – locking rollers (adapted from [58])

For preliminary review

Electric blowers are used in combined supercharging systems of low-speed engines in start-up and light load modes.

Radial-axial type fans are used as air blowers (Fig. 4.86).

The blowers turn on automatically when the engine starts or when the boost pressure drops below the set norm. To automatically separate the cavities of the air receiver and the blowers, movable, hinged air dampers are used.

4.10 Engine intake tract elements

Elements of the engine intake tract include air pipes, charge air temperature stabilizers, moisture separators and intake receivers.

Air pipes are metal boxes of round, rectangular or transitional cross-section (for example, with a transition from round to rectangular) through which air is supplied to various elements of the intake tract. Connecting flanges are installed at the ends of the gas ducts (Fig. 4.87).

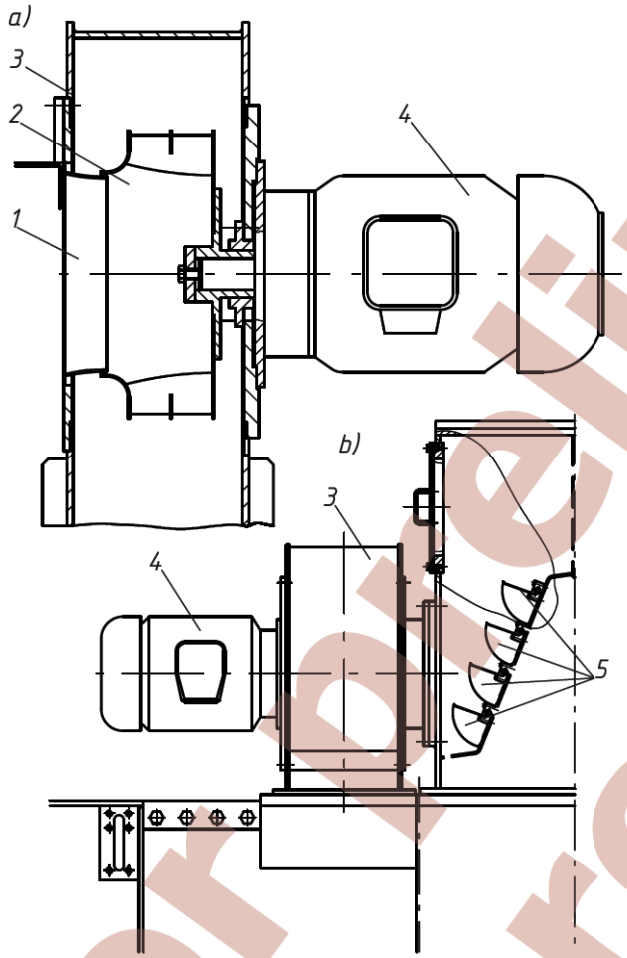


Figure 4.86 – Electric blower (*a*) and its inclusion in the air supply system of a diesel engine type RTA 72 from Wärtsilä (*b*): 1 – inlet deflector; 2 – rotor; 3 – housing; 4 – electric motor; 5 – check valves (adapted from [11])

In some cases, air ducts are equipped with thermal expansion compensators made in the form of metal bellows. If temperatures allow, expansion joints can be made in the form of spacers made of elastic non-metallic composite materials.

Charge air temperature stabilizers. As noted earlier, a side effect of the process of compressing air in a compressor is a significant increase in its temperature. In some cases, its value can reach 160...210°C. If air is supplied to the engine at such temperatures, the weight filling of the cylinders will be less, since the air density decreases with increasing temperature. In addition, the thermal stress of the cylinder-piston group parts will increase and the environmental performance of the engine will deteriorate. In this regard, before air is supplied to the cylinders, it is pre-cooled. A decrease in the charge air temperature for every ten degrees increases the mass of the charge entering the working cylinder by 2...2.5% and leads to a decrease in the average operating cycle temperature at increased boost pressure.

For preliminary review

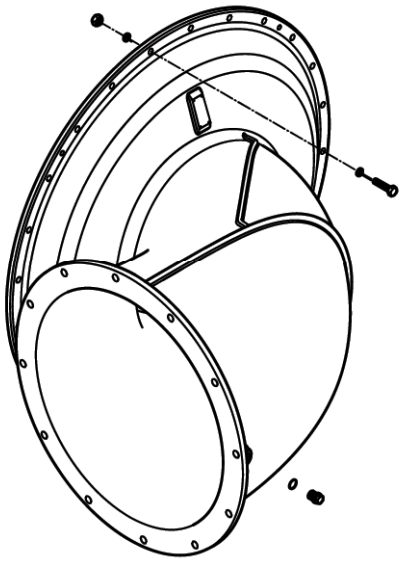


Figure 4.87 – Air supply pipe (adapted from [44])

At low seawater temperatures, cooling of the charge air below the dew point may be accompanied by condensation of moisture vapor. Drops of water can settle on the surfaces of the cylinder liner, as a result of which the lubrication conditions of the parts are disrupted, leading to increased wear. In addition, moisture entering the circulating oil through leaks in the gas joint leads to deterioration of the lubrication conditions of all engine elements. Therefore, usually the temperature at the cooler outlet is maintained 10...15 degrees above the dew point. In this regard, modern engines, and primarily four-stroke ones, are equipped with temperature stabilization systems at the inlet to the intake receiver (Fig. 4.88). These systems include a sequentially installed charge air cooler and heater, connected respectively to the hot water circuit from the engine cooling system and the sea water circuit (Fig. 4.89 *b*). Quantitative redistribution of liquid flows between the circuits makes it possible to stabilize the air temperature at the outlet of the heat exchanger at the level of 40...55°C.

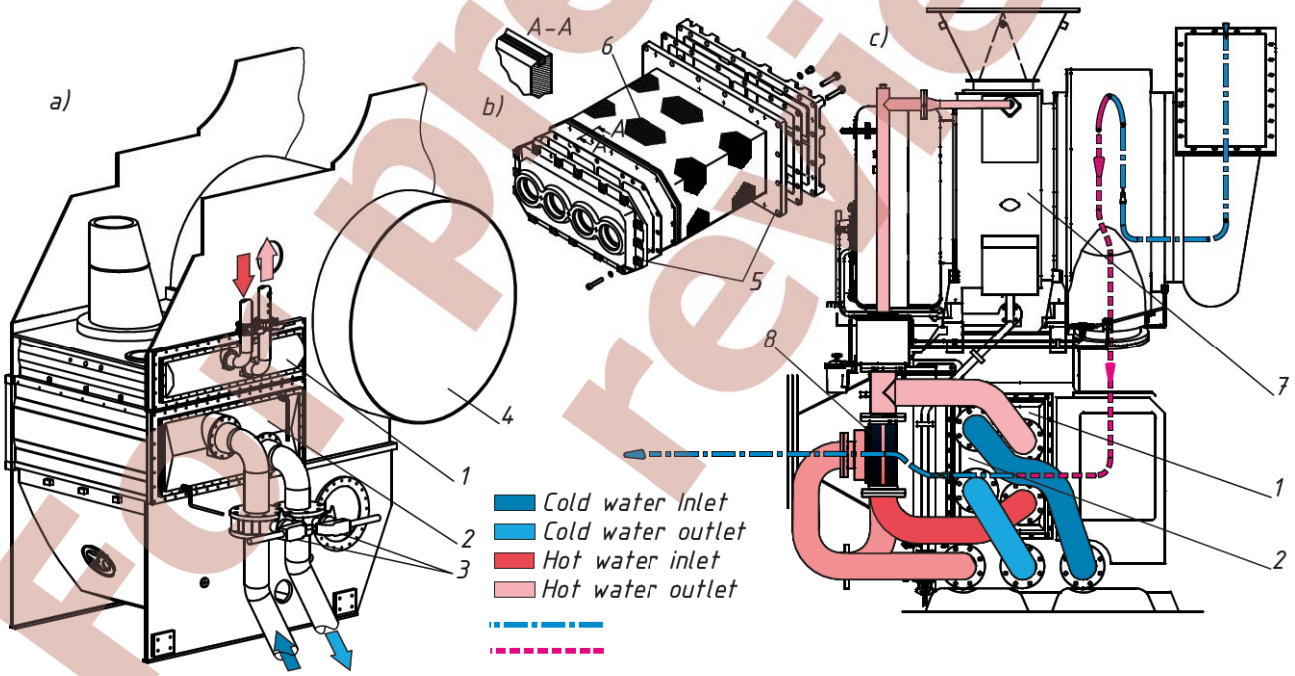


Figure 4.88 – Devices of the charge air temperature stabilization system: *a* – low-speed engines of the MS series from MAN; *b* – heat exchanger of the high-sided engine L 16/24 from MAN; *c* – diagram of the air supply system with charge air temperature stabilizer for the L 46 diesel engine from Wärtsilä. 1 – heater; 2 – cooler; 3 – shut-off valves; 4 – inlet receiver; 5 – support plates; 6 – finned plates; 7 – turbocharger; 8 – charge air temperature regulator (adapted from [12, 19, 27])

For preliminary review

Charge air temperature stabilizers are heat exchangers predominantly of the water-air type, and only in emergency situations Diesel generators sometimes use air- to-air coolers.

In marine engines, three types of heat exchangers are most widespread: tube-band, plate-band and tube-fin.

Tubular-tape heat exchangers (Fig. 4.89 *a*) have several rows of flat tubes made of sheet material, seamlessly drawn or brazed. Corrugated tapes are placed inside and outside the tubes, which serve to increase the heat exchange area and to turbulize liquid or gas flows. The ends of the tubes are placed in the holes of the support plates of the appropriate shape. Tubes, strips and support plates together form the core of the heat exchanger. The tubes are attached to the support plates by soldering or welding. Inlet and outlet tanks equipped with connecting pipes.

Plate-tape ribbed design (Fig. 3.89 *b*) differs from the tubular-tape type in the absence of support plates. In heat exchangers of this type, the channels are closed by prismatic inserts having dimensions corresponding to the heights of the finned strips. The planes of individual inserts along each of the ends of the frame form common planes that serve as support plates. Inlet and outlet tanks are attached to these ends.

Tubular-fin heat exchangers (Fig. 3.89 *c*, *d*) are widely used in pressurization systems of marine internal combustion engines due to their high reliability. The tubes of these heat exchangers are round in shape. During processing on the outside of the tubes form the ribs (Fig. 3.89 *c*).

It is also possible to fin individual tubes or a bundle of them using plates connected to the tubes by soldering (Fig. 3.89 *d*). The ends of the tubes are sealed into the holes of the support plates. Tubes are made from cupronickel alloy or titanium.

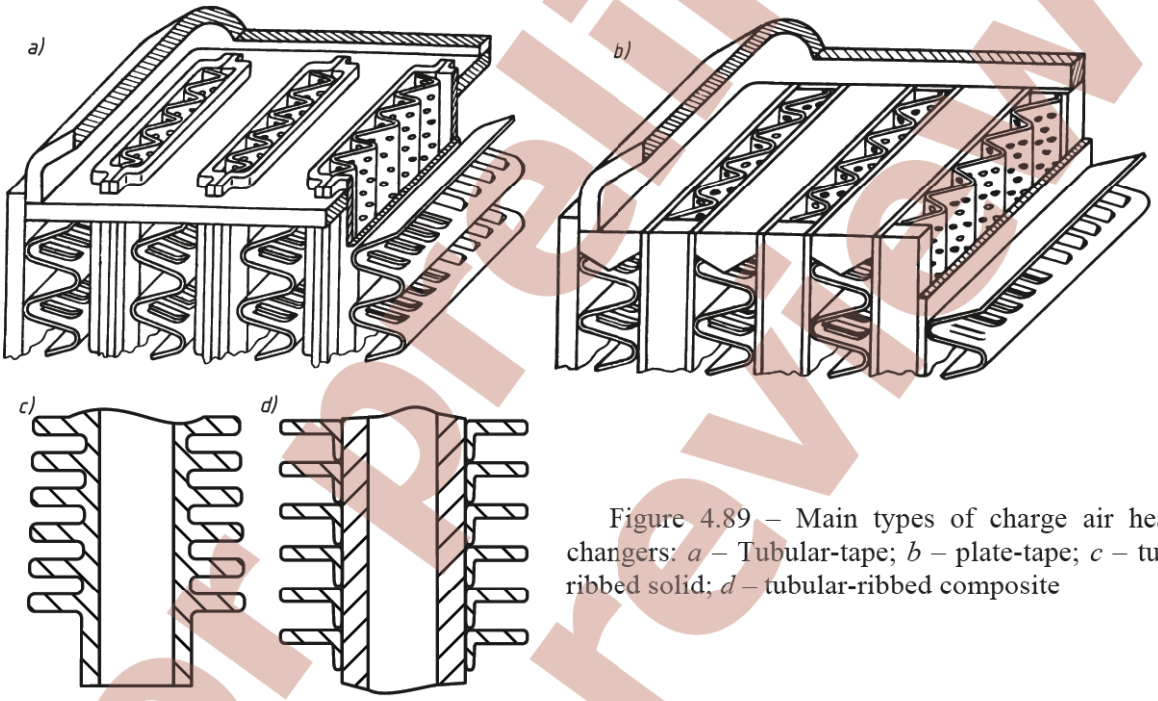


Figure 4.89 – Main types of charge air heat exchangers: *a* – Tubular-tape; *b* – plate-tape; *c* – tubular-ribbed solid; *d* – tubular-ribbed composite

Moisture separators. In most modern engines, to increase their power and efficiency, as well as to reduce the toxicity of exhaust gases, cooling of the charge air below the dew point is used. In this case, to prevent large drops of moisture from entering the working cylinders, the charge air cooler is located below the receiver (Fig. 3.90 *a*), and after the heat exchangers, moisture separators are installed, which are a set of special baffle plates (Fig. 3.90 *b*).

The air, passing through the grilles of profiled plates, changes its direction of movement several times. As a result, moisture particles, which have greater inertia than air molecules, hit the surface of the plates and flow down them into a storage tank. Special pockets formed by pads attached to the plates serve to deflect the air flow from the surface of the plates in order to prevent moisture from being blown off from their surface, creating conditions for its unhindered flow into the assembly tray. A drainage system is provided to remove condensate from the separator.

For preliminary review

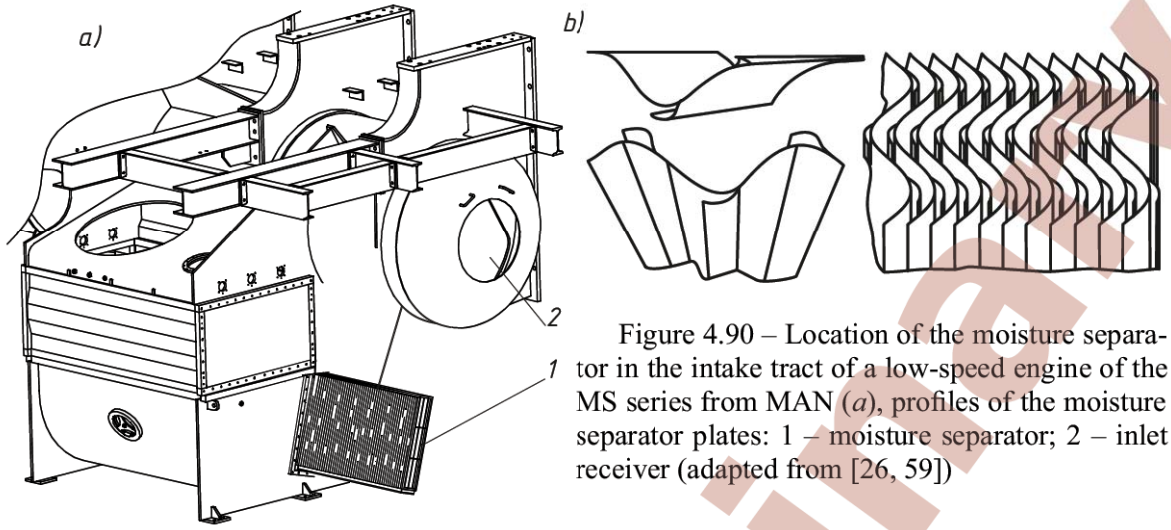


Figure 4.90 – Location of the moisture separator in the intake tract of a low-speed engine of the MS series from MAN (*a*), profiles of the moisture separator plates: 1 – moisture separator; 2 – inlet receiver (adapted from [26, 59])

The intake receiver serves to accumulate air pressure and distribute it evenly across all engine cylinders. In addition, it is an element for reducing noise at the engine inlet. Structurally, the intake receiver can be made as a separate element or integrated into other engine design elements (Fig. 4.91).

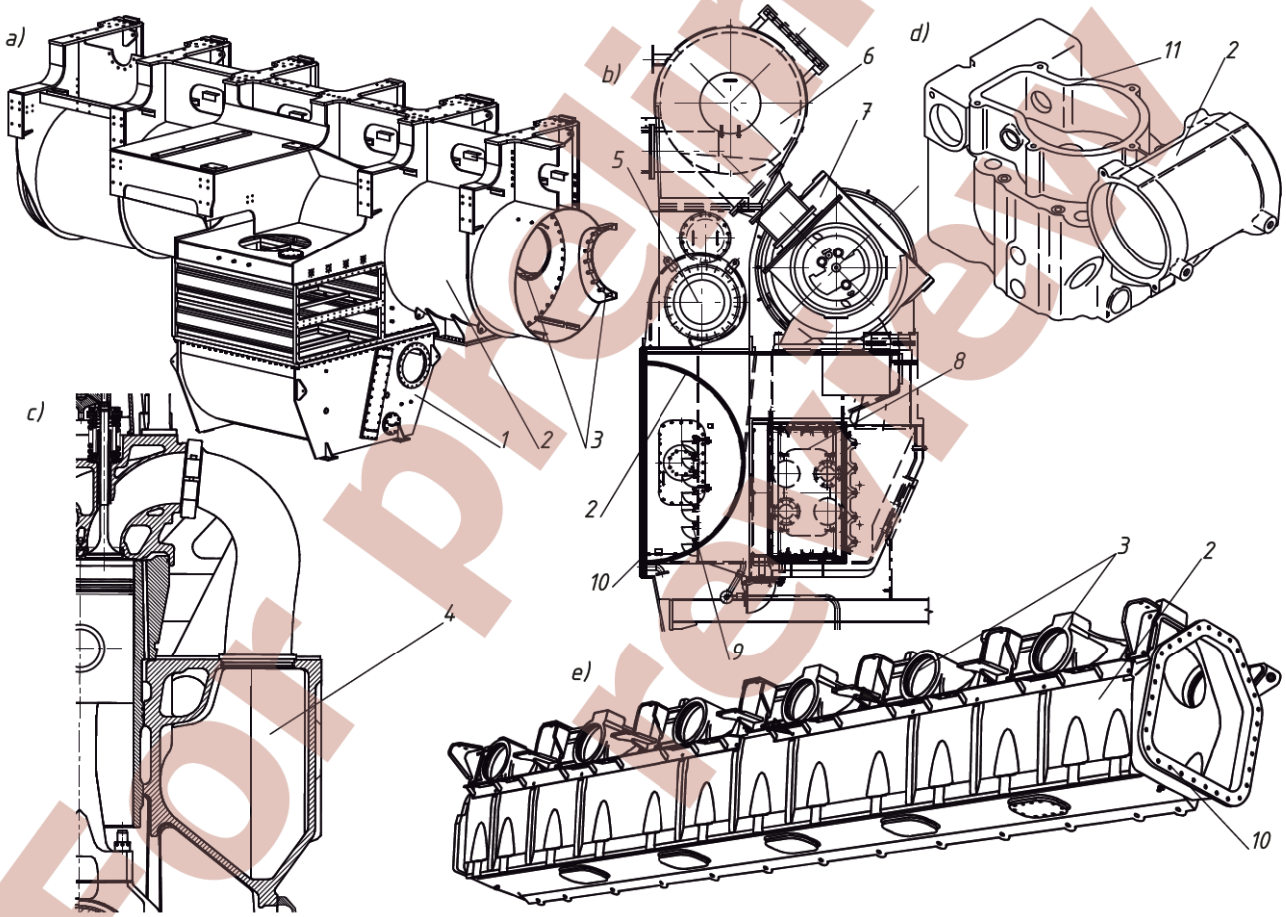


Figure 4.91 – Design of intake manifolds of marine engines: *a* – low-speed engines of the MC series from MAN; *b* – arrangement of elements of the gas-air path of low-speed engines of the RTA series from Wärtsilä; *c* – use of the engine crankcase cavity as an intake receiver in Wärtsilä L46 series engines; *d* – intake receiver made as one piece with the cylinder cover of the L16/24 engine from MAN; *e* – charge air receiver located in the cylinder camber of the V-shaped W46 series from Wärtsilä: 1 – housing of the cooler and moisture separator; 2 – receiver; 3 – connecting pipes; 4 – crankcase cavity used as an air receiver; 5 – electric blower; 6 – exhaust receiver; 7 – turbocharger; 8 – heat exchangers for charge air temperature stabilizer; 9 – hinged dampers; 10 – mating surface of the intake receiver (adapted from [11, 12, 19, 60])

For preliminary review

For example, on low-speed and medium-speed engines, a receiver design made in the form of a cylindrical cylinder closed at the ends with profiled covers has become widespread.

The receiver is connected to the charge air cooler housings by welding, and to connect it to the cylinder jackets, adapter pipes with flanges are welded.

4.11 System elements for exhaust and noise reduction of exhaust gases

The exhaust gas removal, exhaust and noise reduction system consists of an engine exhaust tract, including gas exhaust pipes and exhaust receivers, gas exhaust channels, noise mufflers, exhaust gas toxicity reduction systems and heat recovery systems. The design of the exhaust tract largely depends on the selected engine supercharging scheme, the number of turbocharger units used, and the method of their connection.

As already indicated, on marine engines, charging schemes with pulsed supply of gases to the turbine and with supply of gases to the turbine at constant pressure are used.

To implement each of these charging methods, structurally different gas exhaust tract systems are used in the area from the exhaust channels of the cylinder covers to the entrance to the gas turbine.

In Fig. 4.92 *a* is a diagram of the supercharging of a four-stroke engine with constant pressure in front of the turbine and with the supply of gases to the turbine in the form of pressure pulses (Fig. 4.92 *b*). In the first case, gases from all cylinders enter one common receiver, the volume of which is 10...15 times greater than the volume of the working cylinder. The pressure pulses from the individual cylinders are smoothed out and the gases enter the turbine at a relatively constant pressure.

With pulse charging, the cylinders are grouped in such a way that the pressure pulse in one cylinder does not overlap with the pulse in the other. To do this, individual groups of cylinders are led into separate gas exhaust pipes of a relatively small volume. As a result, gases enter the turbine in the form of pressure pulses through two or more independent channels (Fig. 4.93 *a*).

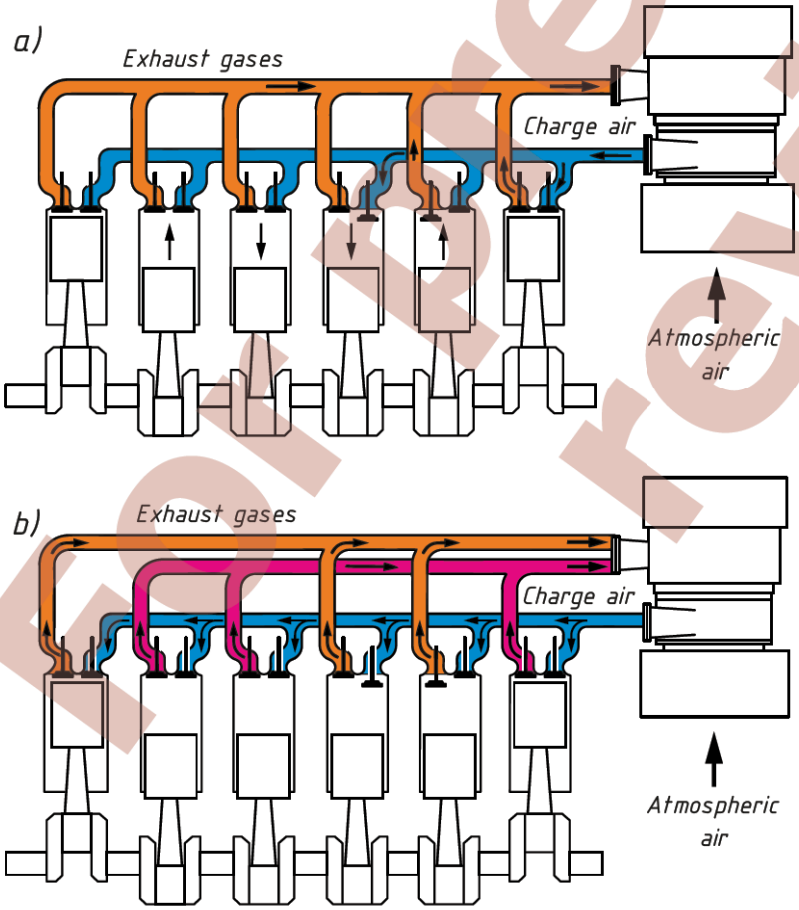


Figure 4.92 – Diagram of the organization of the exhaust tract of a four-stroke engine with constant pressure in front of the turbine (*a*) and with the supply of gases to the turbine in the form of pressure pulses (*b*) (adapted from [61])

For preliminary review

Pulse charging systems are characterized by a high response rate to changes in engine load. Thanks to the small volumes of the cavities between the exhaust valves and the turbine, changes in pressure are quickly transmitted to the latter, and when accelerating, it provides air to the engine in accordance with its new operating mode. This feature is decisive, for example, for diesel generator engines, in which load surges are stepped.

On modern two-stroke engines, the use of high boost has led to the almost complete replacement of pulse systems with systems with constant gas pressure in front of the turbine. This is explained by the fact that when the boost pressure increases above 200 kPa, pulse boost loses its advantages over isobaric boost. Systems with constant gas pressure in front of the turbine are simpler in design, easier to install, their efficiency in the nominal mode is higher and the diesel engine has higher performance than with pulse charging. The transition to such systems made it possible to increase the efficiency of turbochargers by 6...8% and thereby improve air supply in the main operating modes. However, to start the engine and ensure its operation at low load conditions, it is necessary to use additional electric-driven blowers, which turn on automatically when the boost pressure drops below a set value.

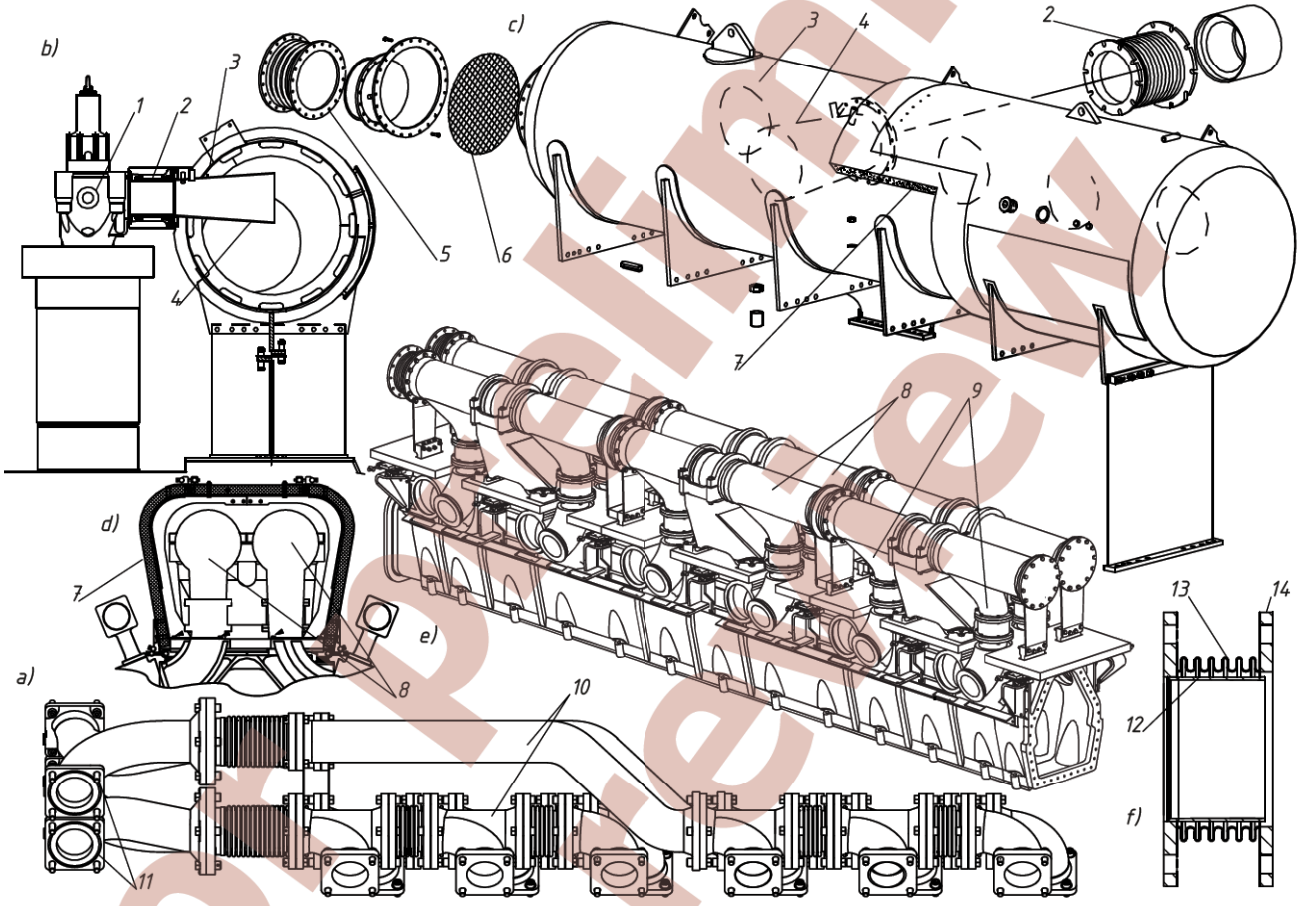


Figure 4.93 – Design of exhaust manifolds of marine engines: *a* – engine with pulse supercharging; *b*, *c* – low-speed engines of the MC series from MAN; *d* – noise and thermal insulation of exhaust manifolds of a V-shaped engine of the W46 series from Wärtsilä; *e* – exhaust manifolds with pulse converters of the W46 series engine from Wärtsilä; *f* – bellows-type thermal expansion compensator. 1 – cylinder cover; 2 – bellows; 3 – exhaust receiver; 4 – bell; 5 – connection pipe for a turbocharger with a thermal expansion compensator; 6 – protective grille; 7 – noise and thermal insulation; 8 – exhaust manifolds with pulse converters; 9 – gas exhaust pipes; 10 – gas ducts; 11 – gas turbine connecting flanges; 12 – inner race of the bellows; 13 – outer corrugated surface; 14 – connecting flanges (adapted from [8, 12, 19, 62])

Structurally, exhaust gas receivers are a metal cylinder that has connecting pipes with which it is connected to the output channels of the cylinder covers (Fig. 4.93 *b*, *c*). In two-stroke engines, to

For preliminary review

reduce exhaust backpressure and reduce exhaust noise, gases are usually discharged through an expanding socket located in the internal cavity of the receiver (Fig. 4.93 *b*).

To reduce energy losses, as well as to reduce noise during operation, exhaust receivers are covered with screens made of heat- and noise-insulating material (Fig. 4.93 *d*). To protect turbines from damage by large objects, protective grilles are installed at the outlet of the receiver.

In modern four-stroke engines, used both as main and auxiliary engines, the exhaust tract is equipped with a pulse converter (Fig. 4.93 *e*), which contribute to the effective conversion of the kinetic energy of gases into the potential energy of their pressure. To do this, gas pipes from individual cylinders enter the main line tangentially. This leads to the fact that the pressure of the gases is equalized as they move along the exhaust tract and they are supplied to the turbine at almost constant pressure. As a result, the efficiency of the turbine increases, and the use of pulse energy increases its power. Thus, this system combines the advantages of pulse and isobaric charging, which allows for a balance of turbine and compressor power at low and medium loads.

To compensate for thermal expansion and absorb vibration vibrations from a pulsating gas flow, special devices called *bellows are installed between the elements* of the gas exhaust system, which are an insert into the pipeline with two connecting flanges (Fig. 4.93 *f*). The outer surface of the bellows gas channel is corrugated. To protect the outer corrugated shell from excessive temperatures, a cylindrical insert is installed inside the channel, acting as a heat shield. In order not to limit the possibilities of axial movements of the corrugated shell, the heat shield is attached to only one of the flanges. This end of the bellows is installed on the gas flow inlet side.

4.12 System elements for reducing exhaust gas toxicity

, the requirements for the content of harmful substances in engine exhaust gases have been significantly tightened, and primarily this concerns nitrogen oxides (NO_x). Significant progress in reducing the content of this group of harmful substances has been achieved by improving the operating parameters of engines. However, in a number of cases, when such a reduction is not enough, the engine exhaust system can be retrofitted with off-cylinder means for reducing the toxicity of exhaust gases.

The most promising method for reducing NO_x content, which has already found practical application in marine internal combustion engines, is selective catalytic exhaust gas purification SCR (Selective Catalytic reduction). Its essence lies in the fact that the processed gases enter a reactor containing a specially selected catalyst (Fig. 4.94). In addition, ammonia gas NH_3 is supplied to the reactor. At a temperature of 350...450°C the following chemical reactions occur in the reactor:

$$4\,NO + 4\,NH_3 + O_2 \rightarrow 4\,N_2 + 6\,H_2O;$$

$$6\,NO_2 + 8\,NH_3 \rightarrow 7\,N_2 + 12\,H_2O,$$

as a result of which nitrogen oxides break down into harmless products – nitrogen and water vapor. The efficiency of the NO_x decomposition process can reach up to 98%.

The reactor is a metal housing filled with catalytic blocks with a large number of parallel channels, the walls of which are coated with an active substance. The design and dimensions of the SCR reactor depend on engine power, exhaust gas temperature, and the desired NO_x reduction level. In the engine room of a ship, installing a toxicity reduction system requires significant space to accommodate the reactor itself, pipelines, spraying and mixing unit and ammonia tank (Fig. 4.95 *a*). For low-speed two-stroke diesel engines, the temperature of the exhaust gases at the turbine outlet is insufficient for decomposition reactions to occur effectively, therefore, reactors are installed directly in front of the turbine. In addition, a system for switching gas flows is needed, which allows, at low load conditions, when the temperature of the exhaust gases is greatly reduced, to supply them directly to the turbocharger turbine.

Examples of various options for installing reactors in the exhaust system of low-speed engines are shown in Fig. 4.95 *b*, *c*, *d*, *e*.

For four-stroke engines, where the exhaust gas temperature in most cases corresponds to the

For preliminary review

specified temperature range, the reactor can be installed immediately at the turbine outlet.

To reduce emissions of other harmful substances, external exhaust gas treatment systems are also used, but consideration of these systems is beyond the scope of this book.

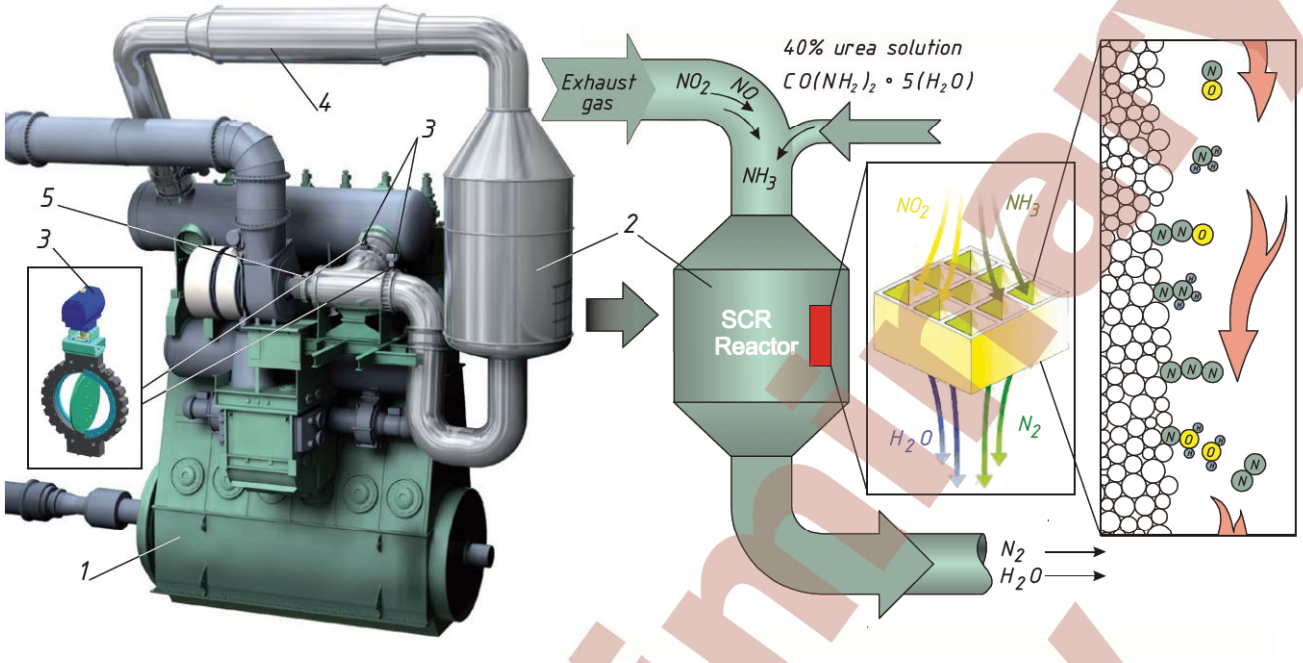


Figure 4.94 – Gas exhaust system of a low-speed diesel engine with an SCR reactor and diagram of the catalytic process. 1 – diesel; 2 – SCR reactor; 3 – gas flow control valve; 4 – spraying and mixing block; 5 – bypass channel (adapted from [63])

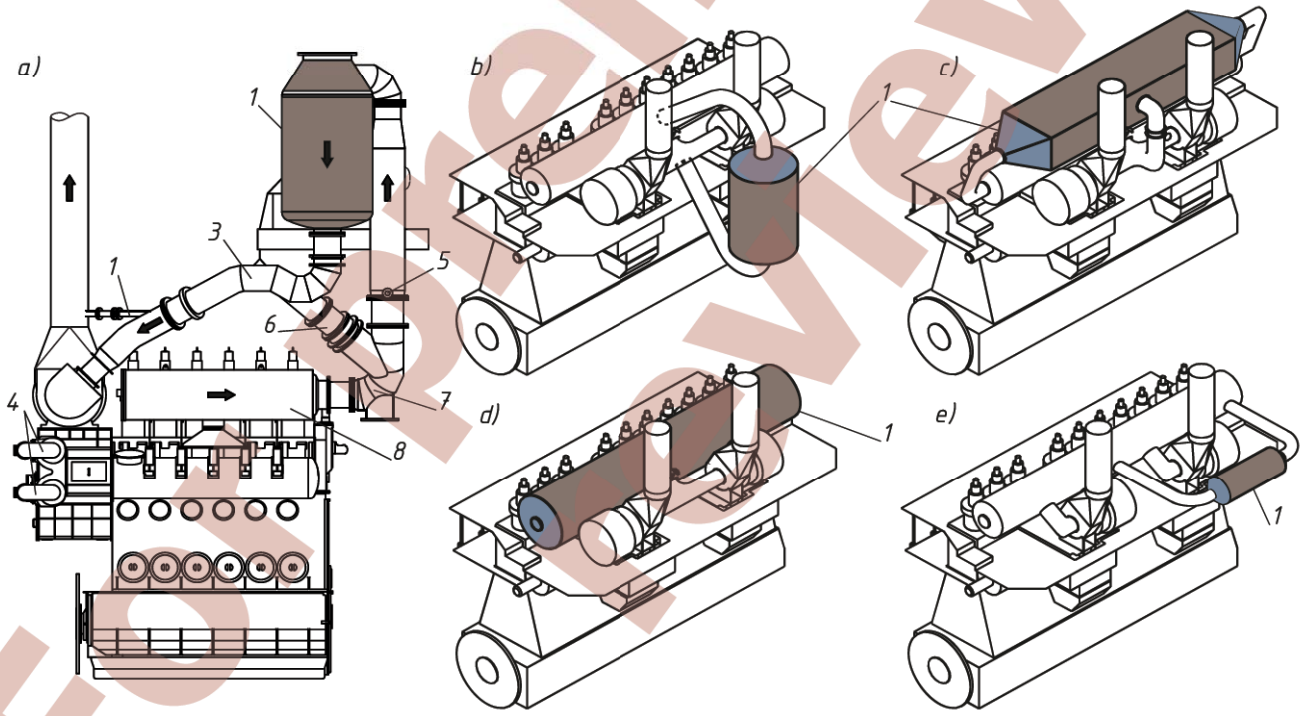


Figure 4.95 – Placement of catalytic reactors in the gas exhaust system of low-speed diesel engines: *a* – gas exhaust system of a 6S35MC low-speed diesel engine with an SCR reactor; *b* – vertical reactor; *c* – horizontal reactor; *d* – reactor built into the diesel exhaust receiver; *e* – reactor for processing part of the exhaust gas flow; 1 – SCR reactor; 2 – turbocharger bypass channel; 3 – temperature control sensor after the reactor; 4 – auxiliary blowers; 5 – introduction of ammonia into the reactor; 6 – reactor bypass channel; 7 – temperature sensor in front of the reactor; 8 – additional flange of the exhaust receiver (adapted from [64])

For preliminary review

4.13 System elements for exhaust heat recovery

The exhaust gas heat recovery system serves to transfer thermal energy to other media for the purpose of its further use or to directly convert heat into mechanical work.

The use of exhaust gas heat to heat other media is usually used for heating purposes. To do this, the gases enter heat exchangers *called recovery boilers,* in which their heat is transferred to water circulating through heated pipes. Typically, such boilers produce water steam, which is supplied for heating fuel, cargo, living and service spaces, as well as for other technological needs of the ship.

The increase in the power of modern engines has led to the fact that, despite a significant reduction in the share of exhaust gas energy in the overall thermal balance, the amount of this energy significantly exceeds the ship's needs for it. In this case, recycling systems are used to convert excess thermal energy into mechanical work.

There are three main schemes for such transformation used in modern shipbuilding:

- part of the exhaust gases is passed past the turbocharger to a recovery power turbine, which can be connected to the engine shaft through a mechanical or electrical transmission. The latest technology is called PTG (Power Turbine Generator) to implement it, an additional electric motor is installed connected to the diesel crankshaft;

- the main part of the exhaust gases after the turbocharger enters the recovery boiler, and some of them enter the boiler past the turbocharger. This makes it possible to increase the temperature of the gases at the inlet to the boiler by 50...60°C and increase its steam production. The steam produced in the boiler is supplied to a steam turbine, which, working on a generator, releases energy into the general ship network or is electrically connected to the engine crankshaft (STG technology (Steam Turbine Generator);

- part of the gases is discharged to the power gas turbine, and after it is supplied to the recovery boiler. This is where the gases come from the turbocharger. The steam produced in the boiler is supplied to a steam turbine, which is mechanically connected to the gas turbine. The energy generated by both turbines is transferred to the engine crankshaft or used to generate electricity (Combined Turbines).

The use of the first technology (PTG) makes it possible to increase its power by 4% when the engine is running at nominal mode. STG technology makes it possible to increase the power of the entire power plant by 5...7% and the combined technology makes it possible to increase the power increase to 10%. At the same time, fuel consumption for modern diesel plants with LSE is 155 g/(kWh).

The power turbines used in such installations are single-stage gas turbines of the same design as the turbines of high-efficiency turbochargers. To remove power from the turbine, a reduction gear is installed on the side of the dismantled compressor wheel.

A more detailed consideration of exhaust gas heat recovery systems is beyond the scope of this book; on this issue, one should refer to specialized literature.

Literature used in writing section 4:

1. Emissions Reduction Technologies. MAN Diesel SE. 2008 – 37 p.
2. Schieman J. ABB Turbocharging. Operating turbochargers – Collection of articles by published in Turbo Magazine 1992-1996. – 35 p.
3. Weisser G. Langsamlaufende Zweitakt-Dieselmotoren. In: Tschöke, H., Mollenhauer, K., Maier, R. (eds) Handbuch Dieselmotoren. Springer NachschlageWissen. Springer Vieweg, Wiesbaden. 2016
4. Белоусов Е.В. Анализ путей развития воздушно-газовых систем судовых двигателей внутреннего сгорания. Матеріали 6-тої Міжнародної науково-практичної конференції «Сучасні енергетичні установки на транспорті і технології та обладнання для їх обслуговування». Херсон. 24-25 вересня 2015 – Р. 18-24
5. Brown D.T. A History of the Sulzer low-speed marine diesel engine. Sulzer Brothers Ltd., Winterthur, Switzerland. 1984. – 47 p.

For preliminary review

6. Woodyard D. Pounder's Marine Diesel Engines and Gas Turbines. Eighth edition. Elsevier Ltd. 2004 – 875 p.
7. Operating Instructions TBD 645. 0299 6391 En, Motoren-Werke Mannheim AG, Mannheim. Germany. – 244 p.
8. MAN Diesel. 50-108ME-ME-C. Vol. I. Operation. MAN B&W Diesel A/S. Copenhagen, Denmark. – 623 p.
9. https://www.clarkipinc.com/emd/cylinder-heads-and-components (14.02.24)
10. High-Powered ADD3OV Medium Speed Diesel Engine / S. Mizuhara, M. Kunimitsu, O. Beppu, M. Takahashi, A. Sakane, M.Tanaka // Technical Papers. 1999 – P. 65-73.
11. RTA72U-B. Operating Manual «Marine». Wärtsilä Switzerland Ltd, Switzerland. 2000. – 277 p.
12. Wärtsilä 46 C. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 421 p.
13. Engine operating instructions. Type M43C. Caterpillar Motoren GmbH & Co. KG. 2007 – 2351 p.
14. Engine operating instructions. Type M25. Caterpillar Motoren GmbH & Co. KG. 2008 – 1481 p.
15. Technical Documentation Engine. Spare Parts Catalogue. Engine 8L 48/60 B. MAN B&W Diesel AG. Augsburg, Germany. 2004 – 339 p.
16. Jakobsen S.B. Service Experience of MAN B&W Two Stroke Diesel Engines. MAN Diesel. Korea – October 2010 – 56 p.
17. Sulzer RTA84C. Operating Manual. «Marine» (UG Version).Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2001 – 297 p.
18. Wärtsilä RT-flex84T and Wärtsilä RTA84T technology review. Wärtsilä Corporation. 2007 – 24 p.
19. Instructions for Main Engine Type S50MC-C. Vol. III Components And Descriptions. MAN B&W Diesel A/S. Copenhagen, Denmark. – 168 p.
20. L23/30H. Section 507 Camshaft and Camshaft Drive. 50705-07H. MAN B&W Diesel A/S. Holeby Generatinc Sets. 1994. – 2 p.
21. Workshop Manual TBD 645. Motoren-Werke Mannheim AG. Mannheim. – 160 p.
22. Bilousov I., Bulgakov M., Savchuk V. Modern Marine Internal Combustion Engines. Springer International Publishing. 2020. – 385 p.
23. LNG Carrier Propulsion by ME-GI Engines and/or Reliquefaction. MAN B&W Diesel A/S, Copenhagen, Denmark. – 26 p.
24. MAN Diesel. MAN Diesel SE. 2008 – 110 p.
25. The Intelligent Engine: Development Status and Prospects. – 19 p.
26. Engine Type S50ME-C8. Volume III Components Description. DS 42014238. STX Dalian Engine Co.,Ltd. – 202 p.
27. Wärtsilä RT-flex96C Operating Manual «Marine» (WECS-9520 / Dynex Servo Oil Pumps). Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2007 – 373 p.
28. Environmental Friendly Two-stroke Marine Diesel Engine: «Mitsubishi UEC Eco-Engine» / Sugihara M., Edo K., Tanida T. // ISME Tokyo. 2005 – 8 p.
29. Service experience. MAN Energy Solutions. Future in the making. MAN Energy Solutions, Denmark. 2019 – 52 p.
30. RT-flex Training. Function of the different Hydraulic- and Pneumatic Components. Chapter 30, Wärtsilä Land & Sea Academy. 2007 – 21 p.
31. Technical Documentation Engine Spare Parts Catalogue. Engine 9L 32/40. MAN B&W Diesel AG. 1996 – 248 p.
32. Hupli J., Troberg M. The smokeless Sulzer ZA40S. Technology, Wärtsilä Italia S.p.A. Marine News – №1 2005 – P. 23-25
33. Technology review Wärtsilä 46F engine. Wärtsilä Corporation. Helsinki. 2004 – 20 p.
34. Engine type ZAV40S. Spare parts catalogue. Wärtsilä Corporation. 2004 – 325 p.
35. Technology Review. Sulzer RTA84T. Wärtsilä Corporation. Helsinki. – 24 p.

For preliminary review

36. ME Engines – the New Generation of Diesel Engines. MAN B&W Diesel, Denmark. 2003 – 21 p.
37. Influence of Ambient Temperature Conditions on Main Engine Operation. MAN B&W Diesel A/S, Copenhagen, Denmark. – 15 p.
38. Technical Documentation Turbocharger Operating Instructions type TCA55-2. MAN Diesel SE. Augsburg. 2010. – 264 p.
39. A century of turbocharging. Turbo Magazine. ABB Turbo Systems Ltd. Baden. Switzerland. 2005 – 28 p.
40. Schieman Ir.J. 5 Years of turbocharging. the diesel engine and turbocharging. ABB Turbo Systems LTD. SWZ. 1994. – P. 520-524.
41. ABB Turbocharging. We multiply power. ABB Turbo Systems Ltd. Baden. Switzerland. 2012 – 20 p.
42. NA/S and NA/T9 The trend setters. MAN Diesel SE. Augsburg, Germany. – 12 p.
43. Operationalarea for turbocharger. MAN B&W Diesel AG. 2005 – 33 p.
44. Operation Manual. Turbocharging TPL76-C33. ABB Turbo Systems Ltd. Baden, Switzerland. 2008 – 132 p.
45. Operation Manual. Turbocharging TPS52-F32. ABB Turbo Systems Ltd. Baden, Switzerland. 2006 – 117 p.
46. Operation Manual. Turbocharging VTR454-11/-21. ABB Turbo Systems Ltd. Baden, Switzerland. 2016 – 280 p.
47. Project Planning Manual NA series. MAN B&W Diesel AG. Augsburg, Germany. 1998 – 174 p.
48. Technical Documentation. Exhaust Gas Turbocharger. Turbocharger TCA 66 – 40037. MAN B&W Diesel. Augsburg, Germany. 2006 – 294 p.
49. Product information. ABB Turbocharging VTR. ABB Turbo Systems Ltd. Baden, Switzerland. 2009. – 8 p.
50. Product information. ABB Turbocharging TPL..-B. ABB Turbo Systems Ltd. Baden, Switzerland. 2009 – 8 p.
51. Latest technologies for environmental measure in 2 stroke diesel engine. WMS-B382. Mitsubishi Heavy Industries Marine Machinery & Engine CO., LTD. 2013 – 24 p.
52. Lassila T. Turbocharging and environmental requirements. ABB Ltd. 2006 – 18 p.
53. VTA Project Guide. Variable turbine area for TCA turbocharger. MAN Diesel & Turbo. Germany. 2009 – 39 p.
54. NR – Project Guide. Turbocharger. MAN Diesel&Turbo. Augsburg, Germany. 1997 – 161 p.
55. Shiraishi K. MET Turbochargers and ORC Generator Value for Environment and Economy. Mitsubishi Heavy Industries Marine Machinery & Engine CO., LTD. 2016 – 37 p.
56. Shiraishi K., Ono Y. Hybrid Turbocharger with Integrated High Speed Motorgenerator. Mitsubishi Heavy Industries, Ltd. Vol. 44 No. 1. 2007 – 3 p.
57. Kawasaki-Green Eco Turbine. Achievement lower fuel consumption for 2-stroke diesel engine by waste heat recovery Simple and high reliable turbo-compound system. Kawasaki Heavy Industries, Ltd. Tokyo, Japan. – 2 p.
58. 710G Series turbocharged power. Products engines. Electro-Motive Diesel, Inc. Illinois, USA. 2006 – 542 p.
59. Barrow S. New technologies bring benefits to old engines. Marine News №2, 2001 – P. 28-32
60. L16/24 Project Guide – Marine Four-stroke GenSet compliant with IMO Tier II. MAN Diesel & Turbo. 2015 – 460 p.
61. Project Guide. Exhaust Gas Turbocharger TCA-Series. MAN Diesel. 2007. – 64 p.
62. Wärtsilä 46. Spare Part Catalogue. Wärtsilä Corporation. – 440 p.
63. Tier III Two-Stroke Technology. MAN Diesel&Turbo. Copenhagen, Denmark. 2012 – 36 p.
64. Emission Control MAN B&W Two-stroke Diesel Engines. MAN B&W Diesel A/S, Copenhagen, Denmark. – 22 p.

For preliminary review

SECTION 5

Fuel systems of marine diesel engines. Mixing and combustion in diesel engines, indicators operation of fuel equipment

5.1 Marine fuels, their composition and physical and chemical properties

Liquid fuels traditionally used for marine diesel engines are products of oil refining and are conventionally made into two main types – light (Marine Diesel Oil – MDO) and heavy (Heavy Fuel Oil – HFO). The basis of all petroleum fuels are predominantly hydrocarbons of the paraffin, aromatic and naphthenic groups. The content of components of a particular group, expressed in fractions of a unit, is called *group composition of the fuel*. Each group of hydrocarbons differs from each other in the number of carbon and hydrogen atoms, the structure of the carbon skeleton and the type of bonds between the atoms, which determine the differences in their properties [1].

Hydrocarbons of the paraffin group are divided into saturated (alkanes) and unsaturated (olefins).

Alkanes (paraffins) are hydrocarbons with the general formula C_nH_{2n+2}, in the molecules of which the carbon atoms are connected to each other by single valence bonds, and the rest of their valences are extremely saturated with hydrogen atoms. Hence the name of alkanes – saturated hydrocarbons. The simplest representative of the homologous series of paraffins is methane – CH_4.

Olefins (alkenes) belong to unsaturated hydrocarbons with the general formula C_nH_{2n}. Their molecule contains one double bond. The first representative of the homologous series is ethylene C_2H_4.

Hydrocarbons of the paraffin group, especially alkanes, have the highest ability to spontaneously ignite.

Aromatic hydrocarbons are hydrocarbons whose molecules contain one or more benzene rings. The general formula is C_nH_{2n-6}. The most important representative of aromatic hydrocarbons is benzene C_6H_6, in the molecule of which six carbon atoms, connected by double valence bonds, form a regular hexagon. This structure of the molecule makes it extremely stable, making these hydrocarbons resistant to spontaneous combustion.

Naphthenic hydrocarbons groups are divided into cycloalkanes and cycloalkenes. The molecules of the former contain cycles of different sizes, the carbon atoms in which are connected to each other only by single valence bonds. General formula: C_nH_{2n}. Cycloalkenes contain one double bond and have the general formula C_nH_{2n-2}.

Naphthenic hydrocarbons, in terms of their ability to self-ignite, occupy an intermediate position between paraffins and aromatic hydrocarbons. The high content of naphthenic hydrocarbons in the fuel composition leads to an increase in its viscosity.

The approximate ratio of various groups of hydrocarbons in the composition of marine fuels is given in table. 5.1.

Table 5.1 – Group chemical composition of marine fuels

Type of fuel	Paraffins	Naphthenes	Aromatic
Marine Diesel Oil	0.30…0.58	0.05…0.15	0.30…0.50
Heavy Fuel Oil	0.05…0.50	0.40…0.70	0.10…0.25

In addition to the group composition, the elemental composition of fuels is distinguished. The approximate ratio of the main chemical elements in the composition of marine fuels is given in Table. 5.2.

For preliminary review

Table 5.2 – Elementary chemical composition of marine fuels

Type of fuel	Carbon (*C*)	Hydrogen (*H*)	Oxygen (*O*)	Sulfur (*S*)
Marine Diesel Oil	0.86…0.88	0.124…0.128	0.002…0.006	–
Heavy Fuel Oil	0.85…0.87	0.121…0125	0.004…0.006	0.004…0.04

It is obvious that the group and elemental chemical compositions significantly affect the ability of the fuel to self-ignite. The more saturated and less aromatic hydrocarbons a fuel contains, the higher its ability to self-ignite, which is assessed by the cetane number (CN).

Cetane number (*Cetane Index*) characterizes the ability of a fuel to self-ignite. To determine it, a mixture consisting of two standard, chemically pure hydrocarbons is used: highly flammable cetane $C_{16}H_{34}$, the ability of which to spontaneously ignite is taken as 100, and difficult to ignite α-methylnaphthalene $C_{10}H_7CH_3$ – an aromatic hydrocarbon, the spontaneous ignition of which is taken as zero. By mixing these components in various ratios, you can obtain a standard mixture with any cetane number from 0 to 100 conventional units. By comparing the self-ignition of the fuel under study with a similar indicator of the reference mixture, its cetane number is determined. In other words, the cetane number of a fuel is an indicator of flammability, numerically equal to the percentage (by volume) of cetane in the reference mixture, which has the same ability to self-ignite as the fuel being tested.

Distillate fuels with a high cetane number (CN = 45...60) are used mainly for high-speed diesel engines, which have significantly less time for the combustion process. For heavy fuels used in low-speed diesel engines, CN = 25...40. Along with engine tests, calculation methods are used to assess the self-ignition of fuels based on an assessment of their group composition. For example, the CCAI (Calculated Carbon Aromaticity Index) method from Shell. The CCAI index is an ignition delay indicator, calculated based on the density and viscosity of the fuel oil.

Next, we will consider other important properties and characteristics of fuels that determine the possibility and scope of their use in marine diesel engines.

Self-ignition temperature (*Ignition Temperature; Autoignition Temperature*) – the minimum temperature at which the fuel ignites spontaneously without an external ignition source. The auto-ignition temperature depends on the group composition of the fuel and the physical characteristics of the charge in the diesel combustion chamber. For marine fuels, at an air pressure at the end of compression of 5.5...16.0 MPa, this temperature lies within 200...250°C.

Flash *point temperature* – the minimum temperature at which a mixture of vapors and air ignites upon contact with an open flame. For marine engines, this temperature should not be less than 60°C when tested in a closed crucible. This indicator characterizes the fuel from the point of view of fire safety; therefore, when storing fuel on a ship, the temperature in the tanks must be at least 10°C below the flash point. Fuels with a flash point below 60°C can only be stored on board in closed metal drums (for example, special DMX gas oil with a low cloud point).

Pour *point temperature* is the lowest temperature at which the fuel will continue to flow when cooled at specified standard temperature conditions. The pour point of distillate fuels lies in the range (-10…-20°C). For heavy fuels it can be quite high (up to +10°C). The pour point determines the possibility of pumping fuel through the pipelines of the ship's fuel system at low temperatures.

Heat of combustion (*Specific Energy*) or calorific value is the amount of heat that is released as a result of 1 kg fuel combustion. In the calculations, the lowest calorific value (Q_l) is accepted, which is less than the total calorific value by the amount of the heat of phase transition of water vapor formed during fuel combustion. For marine fuels of petroleum origin, Q_l lies in the range of 39800...44000 kJ/kg. By default, when calculating the operating process of internal combustion engines, Q_l = 42700 kJ/kg is taken.

Kinematic viscosity is *a* measure of the fluidity of a fuel at a certain temperature. It is defined as the resistance force of two layers of liquid with an area of 1 cm^2, located at a distance 1 cm from each other and moving relative to each other at a speed of 1 cm/s, per unit area. The unit of kinematic viscosity is Stokes, denoted as St (1St). In practice, in the technical specifications for fuel supply, kinematic viscosity is indicated in hundredths of Stokes – centistokes (cSt) (note: 1 cSt = 1 mm^2/s). The viscosity of the fuel decreases as it heats up, so it is indicated at a specific

For preliminary review

temperature value: for light fuels – 20°C or 40°C; for severe 50°C, 80°C and 100°C.

Heating of heavy fuels is widely used to regulate their viscosity under operating conditions, since the efficiency of their use significantly depends on this parameter. The viscosity before the injectors must be within the limits specified by the engine manufacturer to obtain optimal spray pattern characteristics. Viscosity that does not correspond to the optimal value (12…15 cSt) leads to poor combustion, deposit formation and energy loss.

Viscosity is one of the main classification characteristics of fuels. Based on viscosity value, marine fuels are divided into:

- distillate, viscosity in the range of 1.5…6 cSt at 20°C; as well as diesel or medium-viscosity distillate (gas oils), with a viscosity of up to 40 cSt at 50°C;
- engine, or medium-viscous, with a viscosity of up to 120 cSt at 50°C;
- fuel oils or high-viscosity heavy fuels with a viscosity of up to 700 cSt at 50°C and above.

Density – mass of fuel per unit volume (kg/m^3). Density gives an idea of the type of fuel; for light diesel fuels it is 830...890 kg/m^3 at 20°C, and for heavy diesel fuels 990...1010 kg/m^3 at 20°C. Density is used to calculate the amount of fuel taken on board.

Sometimes in practice, instead of density, a relative value such as *specific gravity is used,* which is the ratio of the mass of a given volume of fuel at 15°C(60°F) to the mass of the same volume of pure water at the same temperature.

5.1.1 Impurities in fuel

Mechanical impurities in fuel (*sediment*). During the refining of unrefined oil, a small part of the impurities contained in it passes into the final products, as well as impurities that got into the fuel as a result of technological processes (for example, fragments of catalysts) and storage (for example, rust, scale, dust, sand, etc.). The content of mechanical impurities affects the quality of operation and service life of fuel equipment. In diesel fuels, the content of mechanical impurities is not permissible, and in heavy ones, their content should not exceed 0.1...0.3%. In addition, the size of solid particles must be smaller than the gaps in precision vapors of fuel equipment, that is, no more than 4...7 μm.

Aluminium Al and silicon Si can be contained in heavy fuels produced using aluminosilicate catalysts. The ratio between these two components can vary significantly depending on the characteristics of the technological process. In practice, the total content of catalytic fines *Al* and *Si* in the fuel is determined.

Vanadium (*Va*) and *sodium* (*Na*) *content.* An increased content of these elements leads to high-temperature corrosion of the surfaces of the combustion chamber and exhaust tract components.

Ash content (*Ash*). Ash content is a measure of the metals present in the fuel, either as inherent in the fuel or as contaminants. The ash content of light fuels is 0.01...0.02%, heavy fuels – up to 0.2% by weight.

Water in fuel (*Water Content*) reduces its calorific value, firstly because it reduces the mass of combustible components per unit volume, and secondly, part of the heat released from combustion is spent on heating and evaporation of water. The presence of water promotes the activation of electrochemical and chemical corrosion of fuel equipment elements. The ingress of seawater containing sodium salts (*Na*) into the fuel leads to reactions with compounds (*Va*) contained in heavy fuels, which contributes to the formation of melts that cause high-temperature corrosion of engine exhaust parts. In light fuels, water content is not allowed, and in heavy fuels it should not exceed 1.5%.

Sulfur content is an important indicator of fuel quality. The sulfur content of marine fuel depends on the origin of the unrefined oil and its refining process. When fuel burns, sulfur forms sulfur dioxide SO_2 and sulfur dioxide SO_3 anhydrides, which enter the engine sub-piston space along with crankcase gases. When dissolved in water, they form acids that cause low-temperature corrosion of parts of the cylinder-piston group and exhaust channels. Low-temperature corrosion is especially activated when fuel is watered and the temperature of the cylinder liner walls is low, which can lead to condensation of water vapor contained in the scavenging air and exhaust gases on its working surface.

To reduce sulfur corrosion, special oils with a high alkali content are used. To avoid water con-

For preliminary review

densation on the walls of the cylinder liner, it is necessary to maintain its temperature and the temperature of the scavenging air above the dew point for a given engine operating mode.

In recent years, stricter environmental regulations have been aimed at reducing the sulfur content in marine fuels. Thus, on January 1, 2020, the requirements of the International Maritime Organization came into force, establishing the sulfur content (volume) in marine fuels of no more than 0.5% used outside special control zones. Such fuels are known as Very Low Sulfur Fuel Oils (VLSFO). In Sulfur Emission Control Areas (SECA), the use of fuel with a sulfur content of no more than 0.1% is permitted. Such fuels are known as ultra-low sulfur fuel oils (ULSFO). The mass fraction of sulfur in fuel must be indicated in bunkering certificates.

Hydrogen sulfide (H_2S) is a highly toxic gas and can be fatal in high concentrations. It is naturally present in unrefined oil and can be formed during the refining process. Residual fuel oils may contain varying levels of H_2S in the liquid phase, and in some cases higher levels of H_2S may be observed in marine distillate fuels. The limit level of H_2S in the liquid phase is determined at a level of no more than 2.0 mg/kg.

Carbon residue. The carbon residue is determined by laboratory testing performed at a reduced excess air ratio. It does not characterize the combustion conditions in the engine, but it does give an idea of the amount of hydrocarbons in the fuel, which have complex combustion characteristics. For light fuels, the residue should not exceed 0.3...0.5%, for heavy fuels – up to 20%.

5.1.2 Performance characteristics of fuels and methods for their control

Stability – the ability of a fuel to resist the formation of sediment, sludge formation and delamination during *storage*. Heavy fuels are mainly prone to precipitation and separation. The reason for the instability of fuels, as a rule, is the presence of asphaltenes in them. Asphaltenes are the highest molecular weight molecules and contain all the organically bound vanadium and most of the nickel, as well as a relatively high percentage of sulfur and nitrogen. Asphaltenes have a predominantly aromatic structure, with the *C* and *H* atoms arranged in ring structures. During the thermal cracking process, asphaltene molecules lose some of their outer structure, causing some of the asphaltenes to clump together and no longer be held suspended in the fuel matrix, resulting in the formation of sludge.

Compatibility – characterizes the ability of a fuel to maintain stability when *mixed* with other fuels. When mixing fuels does not result in coagulation of asphaltenes, they are considered compatible with each other. Fuel suppliers guarantee the stability of the fuel they supply, but cannot be held responsible for compatibility issues with other fuels. Responsibility for possible problems arising as a result of mixing fuel lies with the person who made the decision to mix fuel.

Total hot filtration sediment must be monitored for all DMB/DFB fuels that do not meet visual inspection requirements (must be bright and transparent). The reason for the formation of organic sediment is the instability of asphaltene molecules formed during the thermal cracking of oil during the production of these types of fuel.

Cloud Point: Cloud point is defined as the temperature at which wax crystals begin to form in the fuel, causing it to become cloudy.

The cold filter plug point is the lowest temperature at which a given volume of fuel drawn under vacuum through a standard filter for a given time will still flow. The use of distillate fuels with high wax content may result in low temperature flow problems as wax may precipitate on the filter element.

Lubricity is the ability to reduce friction between solid surfaces as they move in relative motion. The fuel protects the moving parts of fuel pumps and injectors from wear. To avoid excessive wear, it must have some minimum level of lubricity. Lubricity-improving compounds are naturally present in petroleum fuels, but their amounts can be significantly reduced by processes designed to reduce sulfur and aromatic compounds. This may lead to deterioration of the fuel's lubricity. The use of such fuels increases wear on the fuel pump and injectors. To monitor the lubricity of distillate marine fuels with a sulfur content of less than 500 min^{-1}, the Wear Scar Diameter (WSD) method is used.

For preliminary review

Fuel additives serve to improve their individual performance. They are divided into anti-carbon, anti-corrosion, reducing the pour point, increasing the cetane number, preventing the formation of sediment, etc. Complex multifunctional additives are also used. The disadvantage of additives is their tendency to precipitate during long-term storage, especially when the fuel becomes water-logged.

5.1.3 Marine fuel grades

Marine fuels are designated by a code consisting of a group of letters and numbers. This code is formed from the abbreviation «ISO» (International Organization for Standardization), then the letter «F» denoting fuel (Fuel) and then the designation of the fuel brand, consisting of three letters. The first letter is «D» for distillate fuels or «R» for residual fuels. The second letter «M» denotes «Marine» use of fuel (for fuels containing fatty acids (FAME), the letter «M» is replaced by «F»). The third letter, for example «A», does not carry a semantic load, but indicates a specific brand of fuel. The figure corresponds to the maximum kinematic viscosity of the fuel in cSt at 50°C. Brands can be designated in full or abbreviated form, for example: ISO-F-RMG 380 or RMG 380 [1].

Distillate fuels. According to the ISO 8217:2017 standard, distillate marine fuels of class DM (distillate marine fuels) are divided into seven grades – table. 5.3.

Table 5.3 – Brands of marine distillate fuels and requirements for them

Characteristic		Dimension	Limits	Fuel grade ISO-F-						
				DMX	DMA	DFA	DMZ	DFZ	DMB	DFB
Kinematic viscosity at 40°C		mm^2/s	max.	5.50	6.00		6.00		11.00	
			min.	1.40	2.00		3.00		2.00	
Density at15°C		kg/m^3	max.	-	890.0		890.0		900.0	
Cetane number		-	min.	45	40		40		35	
Sulfur		wt.	max.	1.0	1.0		1.0		1.5	
Flash point		°C	min.	43.0	60.0		60.0		60.0	
Hydrogen sulfide		mg/kg	max.	2.00	2.00		2.00		2.00	
Acid number		mg KOH/g	max.	0.5	0.5		0.5		0.5	
Total hot filtration sediment		mass, %	max.	–	–		–		0.10	
Oxidation resistance		g/m^3	max.	25	25		25		25*	
Fatty acid methyl ester (FAME)		volume, %	max.	–	–	7.0	–	7.0	–	7.0
Carbon residue in 10% volume of distillation residue		mass, %	max.	0.30	0.30		0.30		–	
Carbon residue – micromethod		mass, %	max.	–	–		–		0.30	
Cloud point	in winter	°C	max.	-16	report		report		–	
	in summer	°C	max.	-16	–		–		–	
Cold filter clogging point	in winter	°C	max.	–	report		report		–	
	in summer	°C	max.	–	–		–		–	
Pour point (upper)	in winter	°C	max.	–	-6		-6		0	
	in summer	°C	max.	–	0		0		–	
Appearance		–	–	Transparent and bright					–	
Water		volume, %	max.	–					0.30[s]	
Ash		mass, %	max.	0.01	0.01		0.01		0.01	
Lubricity, (WSD) at 60°C		µm	max.	520	520		520		520	

DMX brand fuel is a pure distillate fuel with a flash point of less than 60° and a low cloud point. Mainly used for emergency diesel generators. Must be stored outside the engine room in closed tanks.

DMA brand fuel is a pure distillate fuel commercially known as Marine Gas Oil (MGO).

DFA brand fuel – in 2017, the sixth edition of the ISO 8217 standard introduced additional grades of distillate marine diesel fuel with a maximum Fatty Acid Methyl Ester (FAME) content of

For preliminary review

7.0% by volume. The properties of DFA are close to the DMA brand, but it contains FAME.

DMZ brand fuel is identical in characteristics to DMA brand, but has a higher minimum viscosity.

DFZ brand fuel of the brand is identical in its characteristics to the DMZ brand, contains the addition of biodiesel fuel – FAME.

DMB brand fuel is similar in characteristics to DMA grade, but may contain a small amount of residual components. Commercial name: Marine Diesel Oil (MDO).

DFB brand fuel is identical in characteristics to DMB brand, contains the addition of biodiesel fuel – FAME.

Heavy marine fuels RM (residual marine fuels) are the widest group of marine fuels, which are mixtures of heavy residual fractions of oil distillation with distillate fractions (ISO 8217 standard) (Table 5.4).

Table 5.4 – Grades of marine residual fuels and requirements for them

Characteristic		Dimension	Limits	Fuel grade ISO-F-										
				RMA	RMB	RMD	RME	RMG				RMK		
				10	thirty	80	180	180	380	500	700	380	500	700
Kinematic viscosity at 50°C		mm^2/s	max.	10	thirty	80	180	180	380	500	700	380	500	700
Density15°C		kg/m^3	max.	920	960	975	991	991				1010		
CCAI		-	max.	850	960	860	860	870				870		
Sulfur		wt.	max.	Regulatory requirements [1]										
Flash point		°C	min.	60.0	60.0	60.0	60.0	60.0				60.0		
Hydrogen sulfide		mg/kg	max.	2.00	2.00	2.00	2.00	2.00				2.00		
Acid number		mg	max.	25	2.5	2.5	2.5	2.5				25		
Total besieged		volume, %	max.	0.10	0.10	0.10	0.10	0.10				0.10		
Carbon residue – micromethod		volume, %	max.	2.50	10.00	14.00	15.00	18.00				20.0		
Pour point (upper) [2]	in winter	°C	max.	0	0	thirty	thirty	thirty				thirty		
	in summer	°C	max.	6	6	thirty	thirty	thirty				thirty		
Water		volume, %	max.	0.30	0.50	050	0.50	050				0.50		
Ash		volume, %	max.	0.04	0.07	0.07	0.07	0.10				0.15		
Vanadium		mg/kg	max.	50	150	150	150	350				450		
Sodium		mg/kg	max.	50	100	100	50	100				100		
Aluminium + silicon		mg/kg	max.	25	40	40	50	60				60		
Used lubricating oils (ULO): calcium and zinc; or calcium and phosphorus.		mg/kg	-	calcium > 30 and zinc > 15 or calcium > 30 and phosphorus > 15										

[1] The maximum sulfur content is determined in accordance with legal restrictions.

[2] The pour point must be suitable for the intended area of operation of the vessel.

The widespread use of heavy fuels in marine diesel engines is dictated by the fact that their cost is significantly lower than the cost of distillate fuels. However, heavy fuels have increased viscosity, sulfur content, water and mechanical impurities. For their efficient combustion in marine diesel

For preliminary review

engines, the fuel systems of the power plant and the diesel engines themselves must be equipped with means for heating and cleaning fuels from water and mechanical impurities. To neutralize the harmful effects of combustion products from sulfur on parts of the cylinder-piston group, high-alkaline cylinder oils are used. Tightening of environmental regulations in the field of shipping has led to the need to additionally purify heavy fuels from sulfur compounds (desulfitation) or equip a ship's power plant with a system for purifying exhaust gases from combustion products of sulfur SO_x. All this together significantly reduces the economic effect of using heavy fuels. Despite this, heavy fuels continue to be widely used in the fleet, and the volumes of their consumption are constantly increasing, as, however, the technologies for their production are improving.

The basis for the production of heavy marine fuels is fuel oil, which is a residual fraction formed during the distillation of unrefined oil. To achieve different characteristics and quality levels, these residual fuels are blended with lighter fuels such as MGO or marine diesel fuel (MDO). The resulting mixtures are also called intermediate fuels (IFOs) or marine diesel fuels.

Heavy fuels have a high percentage of heavy molecules such as long chain hydrocarbons and long side chain aromatics. MARPOL defines heavy fuel as either a fuel with a density greater than 900 kg/m^3 at, 15°C or a fuel with a kinematic viscosity greater than 180 cSt at 50°C.

Heavy fuels are designated in accordance with their viscosity (for example: IFO 180 and IFO 380 with a viscosity of 180 and 380 cSt, respectively), as well as in accordance with the encryption system adopted for marine fuels (for example: ISO-F-RMG 380 or RMG 380).

Heavy fuels, as a rule, cannot be pumped at a temperature of 20°C, so they must be preheated in ship tanks to a temperature of at least 40°C. At a temperature of 15 °C, heavy fuels have a bituminous consistency.

5.2 Alternative marine fuels

5.2.1 Marine gas fuels

The constant rise in prices for oil and fuels obtained from it, as well as the tightening of requirements related to the environmental performance of marine engines, are forcing an increasing number of manufacturers to look for alternative solutions related to both the search for new fuels and methods for their use in marine diesel engines [2, 3].

The most promising fuels that can simultaneously reduce both the cost of transportation and the amount of harmful emissions are gas fuels of various origins [4, 5].

The most promising gas fuels can be considered natural gas, consisting mainly of methane (CH_4) and petroleum gases, which are mixtures of mainly propane (C_3H_8) and butane (C_4H_{10}).

The use of gas fuel can significantly reduce the amount of harmful emissions in comparison with fuels of petroleum origin – completely eliminate sulfur emissions, radically (by 90%) reduce emissions of nitrogen oxides (NO_x) and significantly (30%) reduce emissions of particulate matter and carbon dioxide (CO_2). The latter is explained by the higher ratio of hydrogen to carbon in gas fuels, which can range from 2.5...4.0 while for liquid fuels this figure does not exceed two. Other advantages of gas fuels include the absence of liquid fractions, which eliminates oil dilution in the area of operation of the piston rings, and the almost complete absence of ash content leads to improved lubrication conditions and increased service life of the circulating oil. As a result, the service life of gas-powered engines can be increased by 1.3...1.5 times.

Initially, the issue of using gases as engine fuel in the fleet arose for gas carriers and oil tankers.

Gas carriers transporting natural gas in a liquefied state (in literature, Liquefied Natural Gas (LNG)) at temperatures below – 160°C in special cryogenic containers constantly evaporate gas in the amount of 0.135...0.15 % of the total cargo volume per day. If we take an LNG tanker with a capacity of cargo tanks as an example 74000 m^3, then approximately 74 m^3. Considering that the expansion coefficient of natural gas during the transition from liquid to gaseous state is about 600:1, then within a day we will have in the form of evaporation to 44400 m^3 gas at atmospheric pressure. Re-liquefaction of gas is a rather expensive procedure, so it is most rational to use gas vapors as fuel for a ship's power plant.

The main factor complicating the use of gas fuels is the possibility of detonation combustion or

For preliminary review

detonation occurring in gas-air mixtures.

Detonation is a combustion mode in which a shock wave propagates through a substance, initiating chemical combustion reactions, which in turn support the movement of the shock wave due to the heat generated during the exothermic reactions initiated by it. The shock wave and the zone of exothermic chemical reactions behind it are called detonation waves, which propagate through the substance at supersonic speed. The cause of detonation is a number of physical and chemical processes occurring during the compression process against the background of increasing temperature of the gas-air mixture. When a certain temperature is reached, called the self- ignition temperature, the reactions acquire a chain character and proceed with a large release of heat. Detonation combustion in the working cylinder is accompanied by the occurrence of wave processes, a significant increase in temperature and pressure, which leads to rapid engine failure.

As a result, when designing gas engines, it is necessary to provide special design and organizational measures to prevent the engine from operating under operating conditions in which detonation may occur. We will consider these measures in more detail in the relevant chapters.

The resistance of gas fuels to detonation is determined by their methane number.

The methane number for gas fuels is calculated as the volumetric content (in percent) of hydrogen (H_2) mixed with methane (CH_4). Pure methane has a methane number of 100 (knock-resistant), and hydrogen has a methane number of 0 (unstable). Thus, if the methane number of a fuel is 65, this means that this fuel has the same knock resistance as a reference mixture consisting of 65% methane and 35% hydrogen.

Since gas fuels in most cases are a mixture of gases, their detonation resistance depends on the ratio of various components in them and the properties of the gases included in the mixture. The main properties that determine the detonation resistance of various gases are given in Table. 5.5.

Table 5.5 – Properties of gases included in gas fuel

Gas	Formula	Boiling point, °C	Self-ignition temperature, °C	Calorific value, MJ/m^3
Methane	CH_4	-161.5	595	39.85
Ethane	C_2H_6	-88.6	515	70.41
Propane	C_3H_8	-42.1	470	101.42
Isobutane	i-C_4H_{10}	-11.7	462	134.03
Normal butane	n-C_4H_{10}	-0.5	365	133.03

Unlike natural gas, propane-butane mixtures can transform into a liquid state at ambient temperatures under pressure above 1.6 MPa (the so-called liquefied petroleum gases, and in literature Liquefied petroleum gas (LPG)). This makes it possible to store them on board in containers that are relatively simple in design. Propane-butane mixtures can be used to supply the working cylinder both in a gaseous state and in the form of a liquid injected into the combustion chamber with a special fuel injector.

The accumulation of experience in operating tankers and gas carriers has made it possible to expand the scope of using gas fuel in the engines of other types of ships. Today, the use of natural gas is considered as a promising direction for ships – container ships, passenger ships, car carriers, ferries, etc.

The use of liquefied petroleum gas is considered in relation to technical fleet vessels that operate in coastal waters and are subject to increased environmental requirements. The relatively high density of liquefied petroleum gas, combined with moderate pressure values in the cylinders, has led to their fairly widespread use for coastal vessels.

5.2.2 Other alternative marine fuels

Tankers carrying unrefined oil during bunkering and transportation experience the release of gas fractions from the cargo, consisting of a mixture of hydrocarbon gases (in literature Volatile Or-

For preliminary review

ganic Compounds (VOC)). These gases are mixed with inert gases supplied to the tanks to prevent the possibility of the cargo catching fire. As the inert gas becomes saturated with hydrocarbons, it is replaced, and the gas mixtures removed from the tanks are sent for processing, during which hydrocarbon gases are extracted from them. During the voyage of a 300000 dwt tanker from the Persian Gulf to north-western Europe with a cargo of light Persian oil, the amount of evaporation could reach 0.6% of the total cargo volume. This amount is commensurate with the fuel consumption of the main engine.

In addition to liquefied petroleum gas and VOC, a number of substances are being considered as alternative liquid fuels for use in main marine diesel engines, which in the future can replace traditional fuels (Table 5.6).

Table 5.6 – Properties of the most promising alternative fuels

Fuel	DT	HFO 45	DME	Methanol	Ethanol	Gasoline
Chemical formula	C_8-C_{25}	-	CH_3-O-CH_3	CH_3-OH	C_2H_5-OH	C_4-C_{12}
Carbon fraction (wt.%)	85	-	52.2	38	52	86
Hydrogen fraction (wt.%)	15	-	13	12	13	14
Oxygen fraction (wt.%)	0	-	34.8	50	35	0
Molar mass (kg/kmol)	183	-	46	32	46	114
Density (kg/m^3)	840	982	660	798	794	740
Lower heating value (MJ/kg)	42.7	40.9	22.8	20.1	27.0	-
Boiling temperature (°C at 0.1MPa)	180…360	-	-24.9	65	78	27…245
Vapor pressure (MPa at 20°C)	≤1	-	0.53	0.013	0.0059	0.025…0.045
Critical pressure (MPa)	3.0	-	5.37	8.1	6.3	-
Critical temperature (°C)	435	-	127	239.4	241	-
Kinematic viscosity (cSt at 20°C)	2.5…3.0	-	0.19…0.25	0.74	1.2	0.6
Surface tension (N/m at 20°C)	0.027	-	0.012	0.023	0.022	-
Volume modulus of elasticity (N/mm^2 at 20°C, 2 MPa)	553	-	1.549	823	902	1.300
Cetane number	38…53	-	55	<5	8	-
Octane number	15…25	-	Low	109	109	90…100
Self-ignition temperature in air (°C)	250…450	-	350	470	362	250…460
Heat of vaporization (kJ/kg at 0.1 MPa)	250	-	467	1089	841	375
Minimum ignition energy (MJ at φ =1)	0.23	-	0.33	0.21	0.65	0.8
Stoichiometric air/fuel ratio	14.6	13.5	9	6.5	9.1	14.7
Maximum flame temperature (°C at 0.1 MPa)	2054	-	1780	1890	1920	2030
Flammability Limits (% by volume)	0.5…7.5	-	3.4…18.28	6…36	3…19	1.4…7.6
Flash point (°C)	52	-	-41	12	14	-45

These fuels include; dimethyl ether (DME)) produced from natural gas, coal, or biomass; methyl alcohol or methanol; ethyl alcohol (ethyl alcohol) or ethanol (ethanol); gasoline.

All of these fuels differ significantly from each other in chemical composition and properties, but all of them can be supplied to the working cylinder in the form of a liquid under increased pressure. In this regard, leading manufacturers of marine engines have already begun producing commercial engines capable of running on alternative fuels.

For preliminary review

In an effort to reduce carbon dioxide emissions to zero, the use of ammonia (NH_3) and hydrogen (H_2) as fuel for marine engines is being considered.

It is estimated that shipping emits about 1.05 million tons of carbon dioxide per year and is responsible for almost 3% of greenhouse gas emissions worldwide. Ammonia, produced using renewable energy sources, is set to be one of the main fuels for the future of shipping.

Hydrogen can provide virtually zero greenhouse gas emissions if produced by electrolysis of water using electricity generated from renewable sources. Leading manufacturers are actively working on the use of these two promising types of fuel.

5.3 History of the creation and development of marine diesel fuel systems

The problem of developing efficient systems for supplying fuel to the combustion chamber and atomizing it arose almost simultaneously with the development of Rudolf Diesel's rational engine. Since the engine was originally intended to run on coal dust, the fuel system was designed with this fuel in mind. To supply coal dust to the cylinders, Diesel designed an air system in which fuel was blown into the combustion chamber with a stream of pre-compressed air. The first experiments revealed a number of difficulties associated, first of all, with the dosing and supply of coal dust into the working cylinders, so in further experiments the coal was replaced with lamp kerosene. Successful experiments with this fuel for a long time determined approaches to the design of fuel systems, primarily for marine diesel engines. For a long time, ship engines used the so-called compressor fuel supply system, in which compressed air was used to atomize it [6].

The advantage of this system was the relative simplicity of the design and the absence of high requirements for the manufacturing accuracy of the main elements. The disadvantage is the presence of a bulky air compressor, which consumed 10...15% of the power of the engine itself.

It should be noted that, having abandoned the use of coal dust as fuel, R. Diesel attempted to design a liquid fuel supply system with mechanical atomization, but the low level of technology at that time did not allow satisfactory results to be obtained. For this reason, in his further work the inventor was forced to return to the compressor spray system, which had proven itself well during previous experiments.

Subsequently, many designers made attempts to improve the compressor system in order to reduce its dimensions and improve the quality of fuel atomization. Attempts have been made to use water vapor obtained by recycling the heat of exhaust gases to atomize fuel, and other design solutions have also been used, but no significant progress has been made in this direction.

The first who managed to build a compressorless engine with fuel ignition by compression was the Gustav Trinkler. And although in the engine he developed, fuel was supplied to the combustion chamber using an air fuel injector, the engine did without a compressor. The air to power the injector was taken from the cylinder during the compression stroke and pressed to a higher pressure by a special piston device, which was driven by the camshaft.

Of course, the disadvantages inherent in the compressor fuel supply system pushed many engineers and inventors to search for alternative solutions, especially since acceptable designs were known and tested even before the advent of the diesel engine. Back in 1891, the English company Richards Hornsby & Sons produced plunger pumps with operating principles similar to modern ones. However, to seal the plunger in such pumps, a set of seals was used, which did not allow obtaining high enough pressures necessary for high-quality fuel atomization. Only in 1912, the American Otto Persson proposed abandoning the gland seal system, replacing them with an accurate (precision) fit of the plunger and bushing.

On diesel engines, plunger pumps developed by James McKechnie were first used in 1910 by Vickers. The pumps had a spring drive in which the energy required to move the plunger was stored using a cam mechanism by compressing the spring. At the moment corresponding to the start of the supply, the cam released the spring, which actuated the plunger.

An important aspect of choosing the type of fuel system is the ability to accurately dose a cyclic portion of fuel depending on the engine operating mode.

For preliminary review

Early designs of plunger-type fuel pumps used the principle of varying the stroke of the plunger. In 1886, American inventor John Holland patented a pump that used conical cams to drive the plunger. The axial movement of the cam shaft made it possible to change the stroke of the plunger in the sleeve.

Rudolf Diesel in 1900, when developing a plunger fuel pump, used cyclic flow control using a bypass valve (US patent No. 654140). The moment of valve closing and the beginning of fuel injection were determined by the size of the gap in the drive mechanism. The gap depended on the position of a special wedge spacer inserted into the gap between the end of the valve stem and the spring-loaded stop, which performs a reciprocating movement together with the plunger. The system turned out to be insufficiently reliable and had to be abandoned, but the principle of valve control itself was subsequently significantly improved. Already in 1903, the German engineer Imanuel Lanster received American patent No. 729613 for a high-pressure fuel pump, in which the cyclic flow was regulated using special valves actuated by a plunger. With minor changes, this regulatory principle is used on marine diesel engines to this day.

In 1892, Carl Pieper received German patent No. 66057 for a plunger-type fuel pump, in which the flow was regulated by an inclined edge applied to the plunger. When turning the plunger, the relative position of the edge and the filling hole in the sleeve changed. As a result of this, the value of the active (discharge) stroke of the plunger changed while the full stroke remained unchanged. In fact, it was the first fuel pump with spool control at the start of delivery. In 1895, William Scott received English patent No. 9403 for a similar design, but with regulation at the beginning and end of the supply. Almost unchanged, the spool control principle is still used to this day on most diesel engines produced around the world.

An important problem faced by developers of direct fuel systems was the problem of coking of nozzle tip. This occurred as a result of fuel droplets hanging on the tips of the fuel injectors at the end of the injection process. To avoid this, it was necessary to sharply release the pressure in the fuel line. The first to solve this problem was in 1910 by the American engineer Herbert Kempton, who separated the supra-plunger cavity and the high-pressure pipeline with a check (discharge) valve with a throttling hole, through which part of the fuel returned to the supra-plunger cavity on the return stroke of the plunger. The problem was solved more radically in 1913 by Karl Steinbecker, who proposed using a small cylindrical collar located directly under the delivery valve to unload the fuel line. In its modern form, the pressure relief valve was proposed in 1924 by Axel Danielson, an engineer at the Swedish company Atlas Diesel.

An alternative way to develop fuel systems was to combine a fuel pump and injector in one unit. This made it possible to get rid of many problems associated with the presence of a sufficiently large volume of fuel located in the connecting line between the fuel pump and the injector. The first attempt to combine pump and injector into one unit was made in 1911 by Frederick Lamplough, who received British Patent No. 1517 for devices similar to those in use today.

Commercial use of pump-injectors first began in the United States in 1931 on Winton engines. In 1934, Arthur Fielden received US Patent No. 1981913 for a pump-injector design for General Motors two-stroke diesel engines, which is still in use today.

The design of closed fuel injectors with a needle valve and a jet nozzle, widely used today on all types of marine diesel engines, was proposed in 1910 by the English engineer Frederick Livens. The pin injector, common on split-chamber engines, was first proposed that same year by Peter Bowman.

As a significant advantage of modern fuel systems, the possibility of supplying a cyclic portion of fuel through several successive injections is considered. It should be noted that the very idea of dividing the cyclic supply into separate portions belongs to the French engineer Louis Gaston Sabathé. In 1908, he patented the design of a fuel injector with pneumatic atomization of fuel, which implemented the principle of two sequential supplys (US Patent No. 437651). In Sabathé engines, the first fuel supply was carried out before the piston reached TDC and made it possible to provide heat supply at an almost constant volume (isochoric heat supply). The second portion arrived when the piston was already moving down, providing heat supply at almost constant pressure (isobaric

For preliminary review

heat supply). Thus, in Sabathé engines a cycle with mixed heat supply was implemented, which was previously obtained in engines built by G. Trinkler. Some authors call this cycle the Sabathé cycle. However, if the question of who has priority in a mixed heat supply cycle remains debatable, then the priority in using separate fuel injection certainly belongs to Sabathé [7].

In 1913, Vickers developed an accumulator fuel system in which a multi-piston pump pumped fuel into a volumetric accumulator under a pressure of 35 MPa. From there, the fuel was distributed among the cylinders and injected by mechanical injectors. Later, the English company Doxford used a similar system on its low-speed engines. In modern literature, such fuel injection systems are called «Common rail» and are increasingly being implemented on transport diesel engines. Combined with the ability to control the fuel supply process using controllers based on microprocessor technology, the use of the common rail system significantly expands the capabilities of modern diesel engines.

To control fuel supply processes in marine diesel engines with common rail systems, an electromagnetic drive of control valves or spools is widely used. It should be noted that the idea of controlling the injection process using an electromagnetic solenoid was first proposed by the American engineer Thomas Gaff, who received US Patent No. 1059604 for this idea in 1913.

The year 1922 should be considered an important milestone in the history of the creation of diesel fuel systems. In December of this year, the German company Robert Bosch in Stuttgart began production of fuel equipment for diesel engines. Before this, each engine manufacturer made fuel system units independently, based on its own technical and technological capabilities. Bosch became the first specialized company to launch the production of a wide range of units and elements of diesel fuel systems, including marine ones. Narrow specialization has allowed us to significantly improve the quality of our products, thanks to which the company remains one of the world leaders in the production of fuel system components today.

In addition to Bosch, the world's leading manufacturers of fuel system components today are Sulzer (Switzerland), Brice (Great Britain), Motorpal (Italy), MAN (Germany), Frieman-Mein (Austria), L'Orange (Germany), etc.

Over the past decades, the development of electronics has made it possible, first on small engines, and then on medium- and low-speed diesel engines, to introduce monitoring and control systems based on microprocessor technology. The main points in the development of such systems are the control of fuel supply processes, as well as gas distribution, cylinder lubrication, engine starting and reversing.

The first low-speed electronically controlled engines were introduced into service by Sulzer in 2001 and by MAN in 2003. The goal of introducing electronic control was to optimize the engine operating process, reduce harmful emissions from exhaust gases and reduce specific fuel consumption. The use of electronic systems has made it possible to increase flexibility in controlling the injection advance angle, the supply law, and also made it possible to optimize fuel supply parameters over the entire range of engine operating modes.

Sulzer, whose LSE developments have now been inherited by Wärtsilä, has taken the path of introducing a common rail system on its new series of engines, called RT-Flex. MAN and Mitsubishi have developed fuel supply systems with a hydraulic injection pump drive, which is controlled through a system of solenoid valves from an electronic control unit. MAN engines equipped with such systems are designated ME, and Mitsubishi engines are called UEC Eco-Engine.

In medium- and high-speed engines, along with traditional fuel supply systems, leading manufacturers are widely introducing common rail systems of various modifications.

On high-speed engines (Caterpillar, Cummins, Perkins), pump-injectors with electromagnetic control valves are widely used.

Currently, the fuel system of a marine engine is an integral part of the general ship fuel system, which, in addition to it, includes:

- fuel reception and storage system;
- fuel processing and fuel supply system to engines.

Consideration of the last two systems is beyond the scope of this book; to study them, it is neces-

For preliminary review

sary to refer to specialized literature.

Next, the fuel systems of modern marine diesel engines, their elements and circuit solutions will be examined in detail. It is assumed that the fuel is supplied to the engines after pre-treatment in the form in which it can be further used. Pre-treatment is carried out to remove mechanical impurities and water from the fuel, improve its physical properties, reduce corrosion activity, and increase the fuel's ability to self-ignite and burn. To do this, the fuel undergoes pre-treatment, which consists of settling, separation, filtration, and heating of heavy fuel. In some cases, the fuel preparation process may include chemical, hydrodynamic and other types of influence. Particular attention is paid to pre-treatment when using heavy fuels.

The engine fuel system is a set of devices and assemblies that serve for:

- supply of prepared fuel to the elements of the fuel system;
- dispensing fuel in accordance with the operating mode of the engine and for supplying measured portions of fuel to the cylinders in accordance with the order of their operation;
- supplying fuel to the cylinder at a certain section of the cycle according to a given law;
- distribution of fuel throughout the combustion chamber in accordance with the accepted method of mixture formation.

Of the variety of design schemes for fuel systems on marine diesel engines, direct action systems and accumulator fuel injection systems have been used.

Regardless of the chosen scheme, marine diesel fuel systems consist of high and low pressure circuits.

The low pressure circuit is part of the diesel fuel system, which serves for the final preparation and supply of fuel to the high pressure circuit. It includes fuel lines, pumps, filters, mixers and fuel heaters.

The high-pressure circuit is part of the diesel fuel system, which serves to supply fuel to the combustion chambers of the engine under high pressure.

In direct acting systems, fuel is supplied to the injector during the discharge stroke of the injection pump plunger. In accumulator systems, fuel is injected into the combustion chamber from a special storage device called *an accumulator*, into which fuel is previously pumped under high pressure.

5.4 Mechanical direct injection systems

5.4.1 General information about direct injection systems

According to their design, direct-acting systems can be of divided and non-divided types (pump-injectors).

In a separate version, the fuel pump and injector are connected to each other by a high-pressure fuel line. In water supply systems and some MSEs with small cylinder power, the most widely used are block pumps with an in-line or V-shaped arrangement of high-pressure sections, which are located in one housing (Fig. 5.1 *a*).

For LSE and MSE with high cylinder power, an autonomous injection pump is installed for each cylinder (Fig. 5.1 *b*), which is located as close as possible to the injector. This makes it possible to minimize the length of high-pressure pipelines and thereby reduce the harmful effects of large intermediate volumes in which wave processes can occur that disrupt the fuel supply process.

Pump-injectors are units in which a fuel pump and a fuel injector are mounted in one housing. They are distinguished by their great compactness, good weight and size characteristics, and lower drive costs.

By minimizing the volume of compressed fuel, pump-injectors are able to provide high injection pressures. The almost complete absence of wave processes makes it possible to eliminate the need for a delivery valve. Pump-injectors are characterized by: absence of after-injections, sharp cut-off of supply, small injection lag relative to the injection stroke of the plunger. As a result, pump-fuel injectors are less prone to coking, which significantly increases their service life, the spread of supply advance angles by rotation speed is reduced and the required range of its regulation is reduced.

For preliminary review

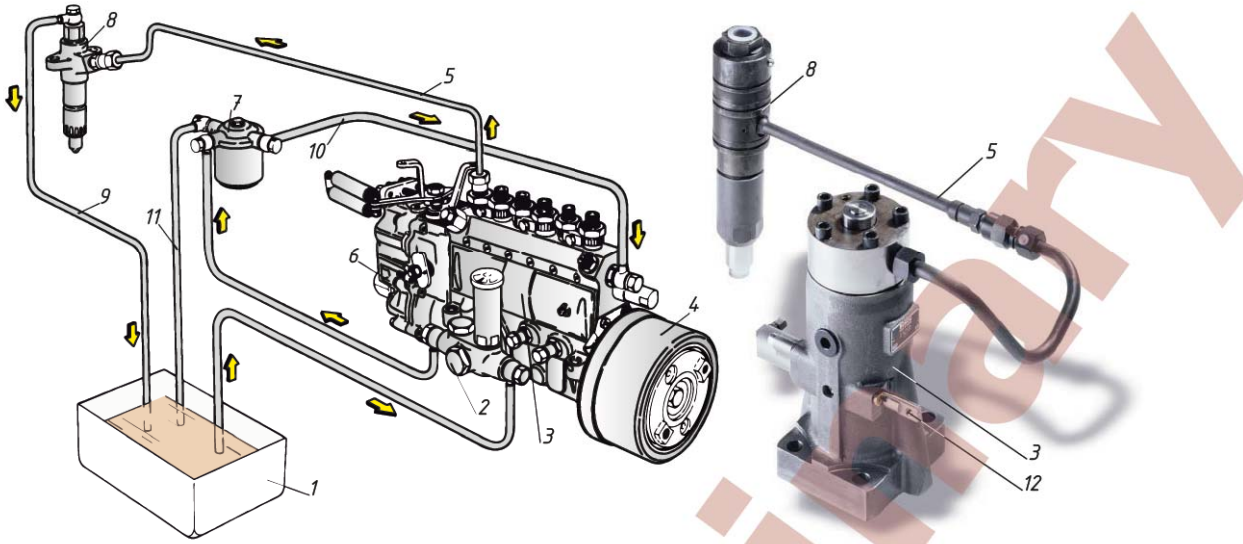


Figure 5.1 – Fuel system with a block-type injection pump (*a*) and with an autonomous injection pump for each working cylinder (*b*). 1 – fuel tank; 2 – low pressure booster pump; 3 – injection pump; 4 – supply advance clutch; 5 – high pressure fuel line; 6 – regulator; 7 – fuel purification filter; 8 – fuel injector; 9 – line for draining fuel leaks from the injector; 10 – fuel drain line from the injection pump; 11 – line for draining unused fuel; 12 – cyclic fuel supply control rail (adapted from [8])

The use of pump-injectors is associated with difficulties in their placement in the cylinder covers and the complexity of the drive, which, as a rule, is carried out from the camshaft through a system of pushers and rods (Fig. 5.2). Currently, pump-injectors are used primarily on high-speed diesel engines of some companies with cylinder diameters up to 300 mm.

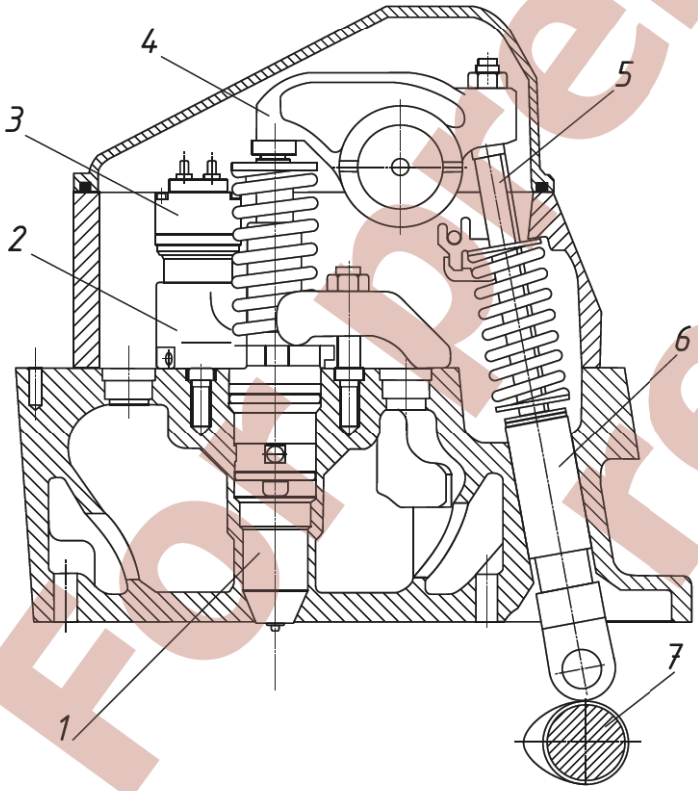


Figure 5.2 – Installation of a pump-injector in the cylinder cover of a high-speed marine diesel engine 3600 series from Caterpillar. 1 – pump-injector; 2 – supply control valve; 3 – solenoid valve; 4 – pump-injector drive rocker arm; 5 – rod; 6 – pusher; 7 – pusher drive cam (adapted from [9])

General diagrams of direct fuel supply systems are shown in Fig. 5.3.

The separate design includes two elements: a high-pressure plunger-type fuel pump and a fuel injector for atomizing fuel, which is connected to the pump by a high-pressure pipeline (Fig. 1.3 *a*). To separate the sub-plunger space and the discharge cavity, a delivery valve is installed between them.

For preliminary review

In fuel systems with a non-divided design (in pump-injectors), both elements of the injection system are combined in one housing, and there is no high-pressure pipeline and delivery valve (Fig. 5.3 *b*).

Schematically, both systems for supplying fuel to diesel cylinders are shown in Fig. 5.3. The plunger 1 of the fuel pump performs a reciprocating movement in the sleeve 2. When it is in the lower (in the upper (Fig. 5.3 *b*)) position, fuel flows through the inlet port 3 into the space above the plunger 4. When moving upward, the plunger closes this port and the fuel, lifting the injection valve 5, enters the high-pressure fuel line 6 to the fuel injector through the formed annular gap between the valve and the valve seat (Fig. 5.3 *a*).

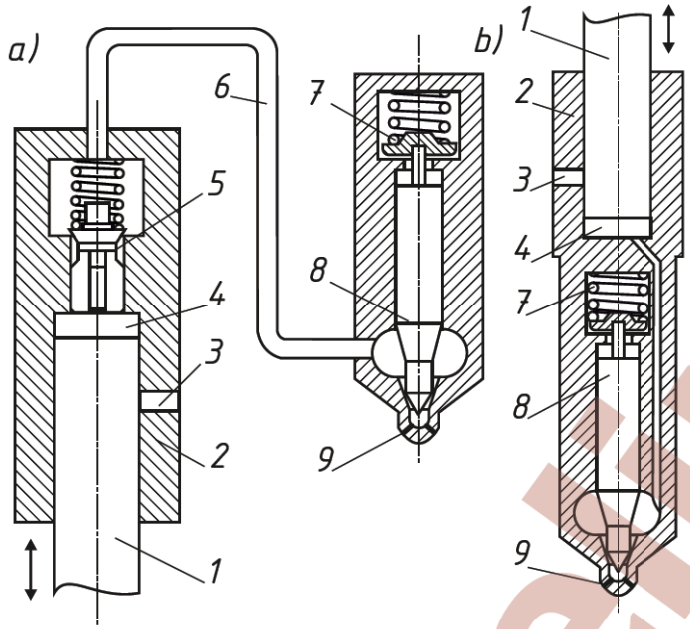


Figure 5.3 – Diagram of the fuel system with separate (*a*) and non-separate (*b*) design of the main elements. 1 – plunger; 2 – plunger bushing; 3 – filling hole; 4 – supra-plunger space; 5 – delivery valve; 6 – high pressure fuel line; 7 – fuel injector needle spring; 8 – fuel injector needle; 9 – nozzle tip (adapted from [9])

Pump-injectors do not have a delivery valve or fuel line, so the fuel immediately enters the sub-needle space of the fuel injector (Fig. 5.3 *b*). When the fuel pressure is on the end surface of the needle the injector 7 reaches a value exceeding the force of the spring 8 pressing it to the seat, the needle will rise and fuel is injected at high speed through the fuel injector of the injector 9 into the combustion chamber. The injection duration, and therefore the cyclic fuel supply, depends on the duration of the active stroke of the plunger, which is the part of the plunger stroke during which fuel is pumped. To change the cyclic fuel supply, various methods of regulating it are used, which will be discussed below.

According to the method of regulating the cyclic supply, high-pressure fuel pumps are divided into spool-type, valve-type and combined. In a number of modern engines, fuel pumps with supply control by changing the full stroke of the plunger have recently found use.

In injection pumps with spool control, the change in fuel supply is carried out by a plunger-spool, and in valve-type pumps – by special valves mechanically driven from a plunger pusher.

Valve pumps have a simpler plunger pair design, but the valve drive system itself has many moving parts, which significantly complicates the pump design. In addition, these pumps are very sensitive to wear of the landing collars in the seats of the suction and shut-off valves, which cause disturbances in the accuracy of injection and an increase in the unevenness of fuel supply to the diesel cylinders.

Spool pumps are simpler in design, easier to regulate and operate. The main disadvantage of spool pumps is more intensive wear of the plunger in the area of the spool edges.

In combined pumps, the cyclic portion of fuel is regulated using both spool and valve control. For example, the beginning of the flow is controlled by a valve, and the end is controlled by a spool thread on the plunger. Such pumps have a rather complex design, so they are practically not used on modern marine diesel engines.

All of the methods discussed above for regulating cyclic flow usually use a fuel injection pump

For preliminary review

with a mechanical drive of plungers (the cam washer is located on the fuel pump shaft or on the diesel camshaft).

Recently, fuel pumps with a hydraulic plunger drive have become widespread on low-speed marine engines. These pumps implement a control principle based on changing the stroke of the plunger, since the plunger does not have a strictly fixed full stroke, determined by the profile of the drive cam. In such pumps, the stroke of the plunger, and therefore the cyclic fuel supply, depends on the amount of oil entering the space under the piston of the hydraulic drive of the plunger. The oil supply from the control line is usually regulated by an electronic control system based on microprocessor technology.

Regardless of the type, the main design differences in the fuel equipment of marine diesel engines are related to the fact that most of them are adapted to operate on viscous IFO, TFO (Intermediate Fuel Oil, Thin Fuel Oil) and high-viscosity fuels HFO, RFO (Heavy Fuel Oil, Residual Fuel Oil). Such fuels, due to their low mobility, cannot be used at ambient temperatures. To reduce their viscosity to 10...14 cSt, before supplying the fuel system units, the fuel is heated to 100...140°C.

Today, almost all manufactured diesel engines are capable of operating on heavy fuels, and only high-speed engines used on ships, mainly to drive emergency diesel generators, use light fuels such as MGO and MDO marine diesel fuel. These fuels do not require heating, since at a temperature characteristic of the ship's engine and boiler room they have a viscosity of 3...12 cSt.

The bulk of marine fuels is a mixture of residual fuel and distillate fractions with a high content of sulfur and other harmful components. In this regard, the fuel equipment of marine diesel engines must be resistant to chemical corrosion.

You can learn more about the properties of fuels used in the marine fleet in specialized literature.

5.4.2 Marine diesel fuel injector

The injectors are designed to inject fuel into the combustion chamber in the form of a finely sprayed aerosol. They must provide optimal conditions for mixture formation, the main of which are fine atomization and uniform distribution of fuel throughout the combustion chamber.

To facilitate the layout on a diesel engine, the injectors should have minimal dimensions. In addition, reducing the volume of the internal cavity of the fuel injector makes it possible to increase the injection pressure and reduce the harmful effects of pressure waves on the fuel supply process. For four-stroke marine medium- and high-speed diesel engines, the central location of the fuel injector in the cylinder cover, when the fuel nozzle tip is equidistant from the walls of the combustion chamber, is the most typical. An example of the central location of an injector in the cylinder cover of a medium-speed MAN L32/40 engine is shown in Fig. 5.4.

For two-stroke diesel engines with a direct-flow valve purge scheme, the presence of a centrally located exhaust valve determined the peripheral location of two or three injectors per working cylinder. In this case, the fuel nozzle tips have strictly oriented holes to cover a certain space of the combustion chamber with jets of atomized fuel.

Of the variety of designs on marine diesel engines, the most widely used are closed-type fuel injectors, that is, fuel injectors in which a special valve is installed that separates the cavity of the fuel injector and the working cylinder during the entire cycle, except for the injection process. As a rule, needle valves with automatic opening under the influence of fuel pressure supplied from the fuel pump to the injector cavity are used for these purposes. Some designs of fuel injectors for medium- and high-speed marine diesel engines are shown in Fig. 5.5.

The needle valve is pressed against the seat using a cylindrical spring, which transmits force to the valve shank through a special movable rod. The presence of a rod allows you to move the spring out of the zone of high temperatures, make it easier to adjust the fuel injector and reduce the size of the nozzle. The rod is made as light as possible, since an increase in the mass of moving parts leads to a decrease in the speed of the needle valve, a delay in the end of injection, and accelerated wear of the locking cone. For these reasons, in some designs they try to avoid a long rod. A number of manufacturers are switching to using fuel injectors with a low spring arrangement and a short rod-plate underneath (Fig. 5.5 *c*). At the same time, the fuel injector itself becomes more compact.

For preliminary review

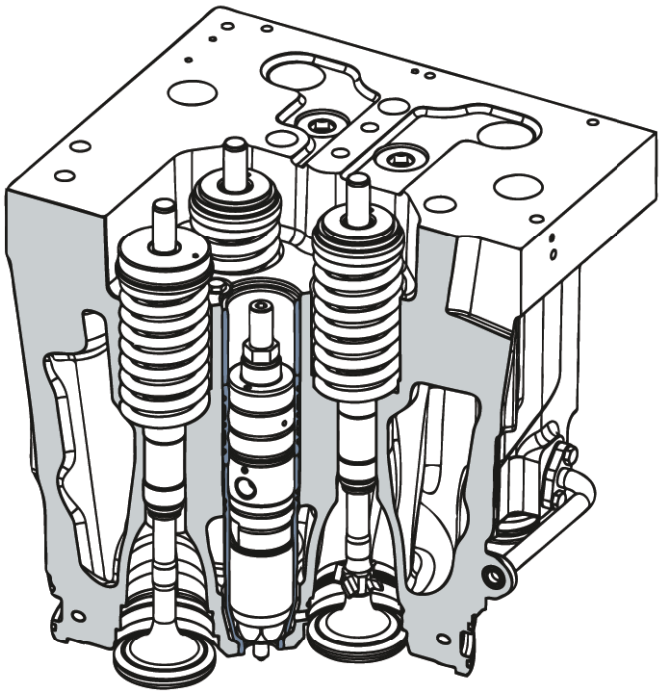


Figure 5.4 – Location of the fuel injector in the cylinder cover of a medium-speed diesel engine L32/40 from MAN (adapted from [10])

The initial tightening of the spring, which determines the opening pressure of the needle valve, is carried out using an adjusting bolt installed at the upper end of the fuel injector, or by installing special calibrated washers under the spring. The latter solution is typical for a number of high-speed diesel engines with relatively low cylinder power.

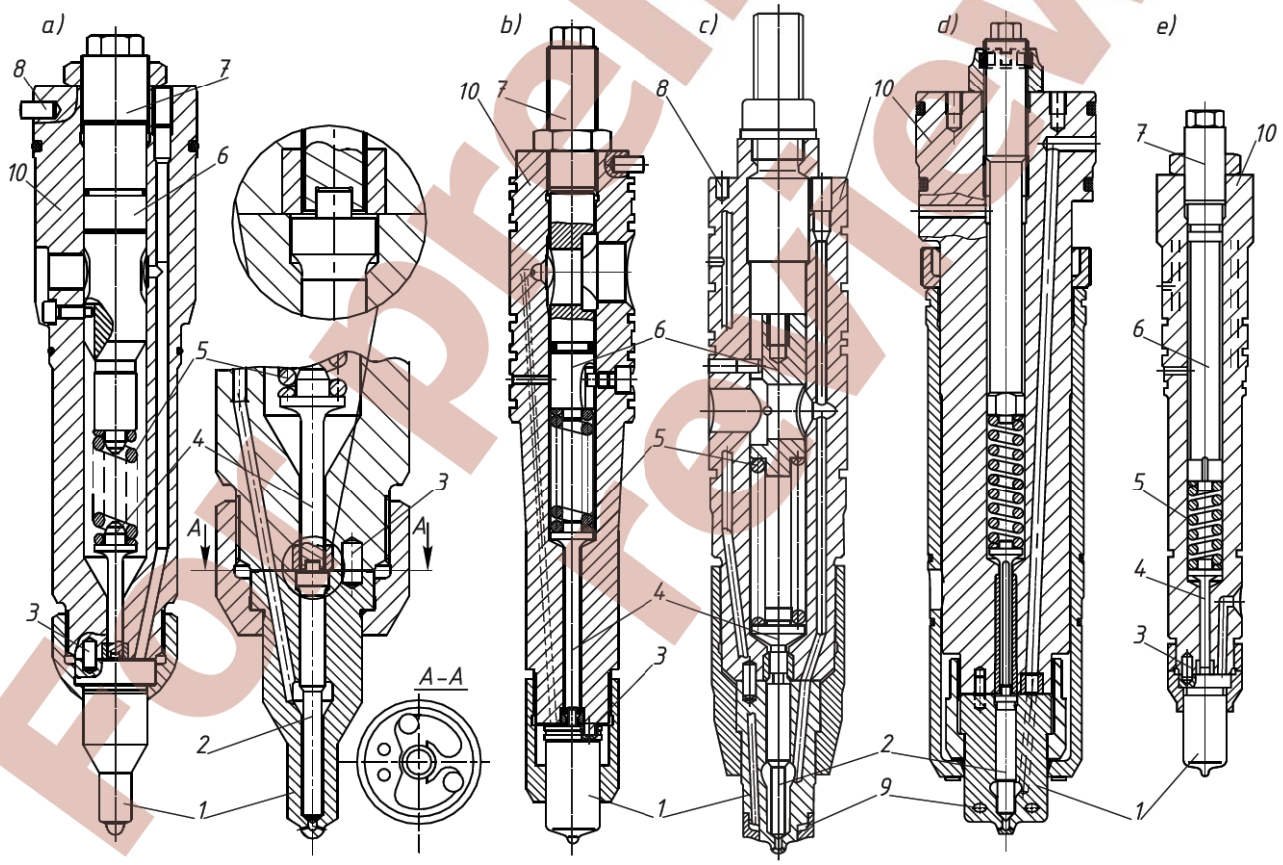


Figure 5.5 – Injectors for medium and high-speed marine diesel engines: *a* – L/W46 from Wärtsilä; *b* – M 25C from MaK; *c* – L32/40 from MAN: *d* – PC2 from MAN-Pielstick; *e* – TBD 645 from Deutz MWM. 1 – spray housing; 2 – needle valve of the nozzle; 3 – mounting pin; 4 – needle valve drive rod; 5 – loading spring; 6 – loading spring stop; 7 – spring pre-tension adjustment screw; 8 – mounting pin; 9 – cavity for coolant; 10 – fuel injector housing (adapted from [10, 11, 12, 13, 14])

For preliminary review

In a number of designs, an intermediate stop is installed between the adjusting bolt and the needle valve spring, in which a groove is made for the fuel supply fitting to the injector to pass through it (Fig. 5.5 *a*, *b*, *c*). The fitting is pressed against the hole on the inside of the wall of the fuel injector housing. The intermediate stop is secured against axial rotation using a locking pin. This solution avoids deformation of the housing under the pressure of the fuel fitting, which can lead to jamming of the injector.

The internal cavity of the chamber for installing the spring is used to collect leaking fuel, which, having leaked along the cylindrical surface of the needle, creates a bath for the spring, ensuring its lubrication and heat removal. This protects the latter from corrosion and reduces dynamic stresses in the turns by 20...25%.

A drain hole for draining leaks into the drainage channel is located in the upper part of the fuel injector to maintain the spring chamber in a constantly filled state.

To prevent leaks from entering the cooling water, special rubber sealing rings are installed on the fuel injector housing, separating the inlet and outlet channels of various media.

The nozzle is the most critical element of the fuel injector design. In Fig. 5.6 shows the designs of some fuel injectors of four-stroke marine diesel engines.

High temperatures operating in the engine combustion chamber can lead to overheating of the nozzle, which can result in jamming of the needle valve, the guide rod of which, together with the guide hole in the nozzle housing, forms a precision pair. As a result of the needle hanging, the nozzle holes become coked. The probability of jamming is especially high for fuel injectors operating on heavy fuels, which are supplied to the fuel injector at a temperature of 100...140°C. When the nozzle overheats, the hardness of the locking surfaces decreases, their wear increases, the size of the gaps in precision connections changes, and their tightness decreases. All this is progressive in nature, as it leads to a deterioration in the operating process conditions in the engine. The maximum permissible temperature of the fuel nozzle tips should usually not exceed 220...240°C, higher values lead to a rapid decrease in their working capacity.

To prevent overheating, cavities are provided in the fuel injector housing for supplying cooling water (Fig. 5.6 *c*, *d*, *e*) or oil (Fig. 5.6 *b*). These same cavities, as well as the internal channels for supplying coolant, make it possible to maintain the temperature regime of the injector when the engine is not running and is in hot reserve.

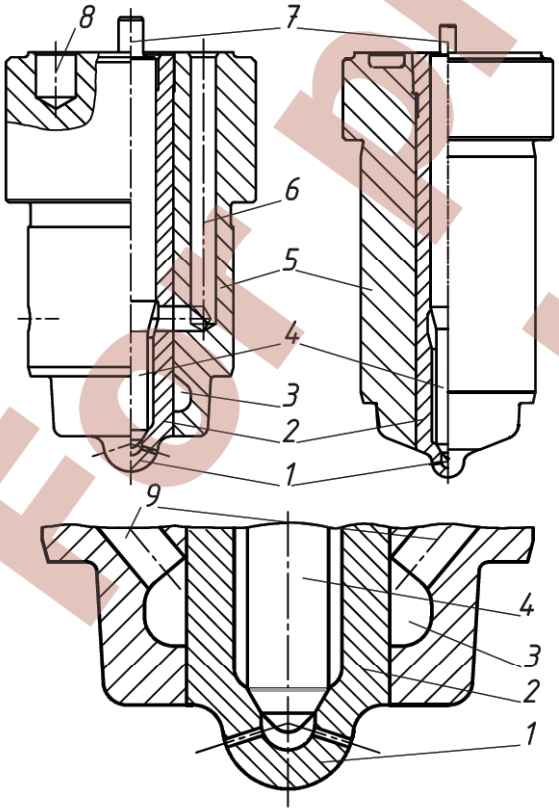


Figure 5.6 – Jet type nozzles: 1 – tip of the nozzle with holes; 2 – guide insert of the needle valve; 3 – cavity for coolant; 4 – needle valve; 5 – spray housing; 6 – channel for supplying fuel to the cavity of the nozzles; 7 – needle valve shank; 8 – hole for the mounting pin; 9 – coolant supply channel (adapted from [15])

For preliminary review

Another option to prevent overheating of the nozzle is based on reducing the area of the protruding part of the nozzle into the combustion chamber. A number of designs of MSE and HSE fuel injectors use long-housing fuel injectors (Fig. 5.5 *a*) with an elongated lower part between the cylindrical surface and the shut-off cone. Their use allows you to remove the precision pair from the most heated lower part and shorten the rod, reducing the diameter of its lower part. In addition, efficient and uniform cooling of the fuel needle and fuel injector housing in the annular gap at the bottom of the needle is ensured. Thermal protection is based on the principle of counterflow: heat flow is directed upward, cold fuel – downward. Reducing the surface area protruding into the combustion chamber leads to the fact that the amount of heat perceived by the nozzle is reduced. In addition, the spray housing itself fits tightly to a brass cup installed in the cylinder cover, intensively cooled with water from the engine cooling system.

For engines with direct fuel injection into the combustion chamber, valve-type multi-jet fuel injectors are widely used. The fuel injector housing can be either solid or composite (Fig. 5.6), with or without a cooling cavity. The lower part of the fuel nozzle is a tip, which has a number of holes drilled at a certain angle to the axis of the working cylinder. The number of holes can be from 1 to 10, and their diameter ranges from 0.12 to 1.05 mm. When the fuel injector is in a central position, the holes are positioned symmetrically around the circumference. If the fuel injector is offset or tilted, the holes are positioned asymmetrically. To reduce fuel leakage, they try to minimize the volume of the sub-needle space, and holes are drilled directly under the shut-off cone. In this case, coking of the nozzle is reduced, soot emissions and, especially, hydrocarbons are reduced.

The fuel nozzle tip is separated from the main volume of the fuel injector cavity by a needle valve, tightly pressed to the seat using a spring. Typically, the angle of the locking cone is 60°, and the angle of the seating surface is 0.5...1.5° less than that of the needle. This ensures quick seating of the needle without a strong impact on the seat and a reliable seal.

As the pressure in the fuel injector cavity increases, the force acting on the valve increases, and when a value sufficient to overcome the spring force is reached, the valve opens, passing fuel into the fuel nozzle tip, and then through the nozzle holes into the engine combustion chamber. The travel of the needle valve is limited by a special stop in the fuel injector housing. The valve lift height usually lies in the range of 0.5...1.5 mm and depends on the size of the injector and the amount of fuel it passes. As the needle stroke increases, the dynamic forces of its impact on the seat and stop increase, which leads to the appearance of work hardening and a loss of needle tightness. To increase service life, the stop is made in the form of an insert into the housing, made of a harder material (Fig. 5.5 *a*).

When the pressure in the fuel injector cavity decreases, the valve needle is lowered under the action of a spring, disconnecting the fuel nozzle tip cavity from the injector cavity. The closing pressure of a needle valve is always lower than the opening pressure. This is explained by the fact that from the moment the needle comes off the seat, fuel pressure acts on the entire surface of the valve. As a result, the pressure required to hold the valve open will be less than the pressure required to open it. This phenomenon is called the differential needle effect. Its presence leads to the fact that the quality of fuel atomization in the final phase will be worse than in the initial stage.

In terms of their principle of operation and a number of design solutions, the injectors of marine low-speed two-stroke diesel engines are similar to the injectors of medium- and high-speed engines, but they have a number of features caused by specific requirements. In Fig. 5.7 shows a general view of the injectors of marine low-speed diesel engines from two leading manufacturers – Wärtsilä and MAN. In Fig. 5.7 *c* shows the design of the engine fuel injector– RT-flex 50B with electronic injection control. This fuel injector has a number of design features that will be discussed below.

The main distinctive feature of low-speed diesel injectors is the widespread use of replaceable spray tips (Fig. 5.7, 5.8). This is relevant for engines, especially when running on fuel oil, as it allows you to replace the fuel nozzle tip, which has the shortest service life, without replacing an expensive precision pair of needle valves. With this design, the tip is pulled together through the nozzle housing with the needle valve housing, forming an additional precision connector (Fig. 5.8 *a*, *d*, *e*).

For preliminary review

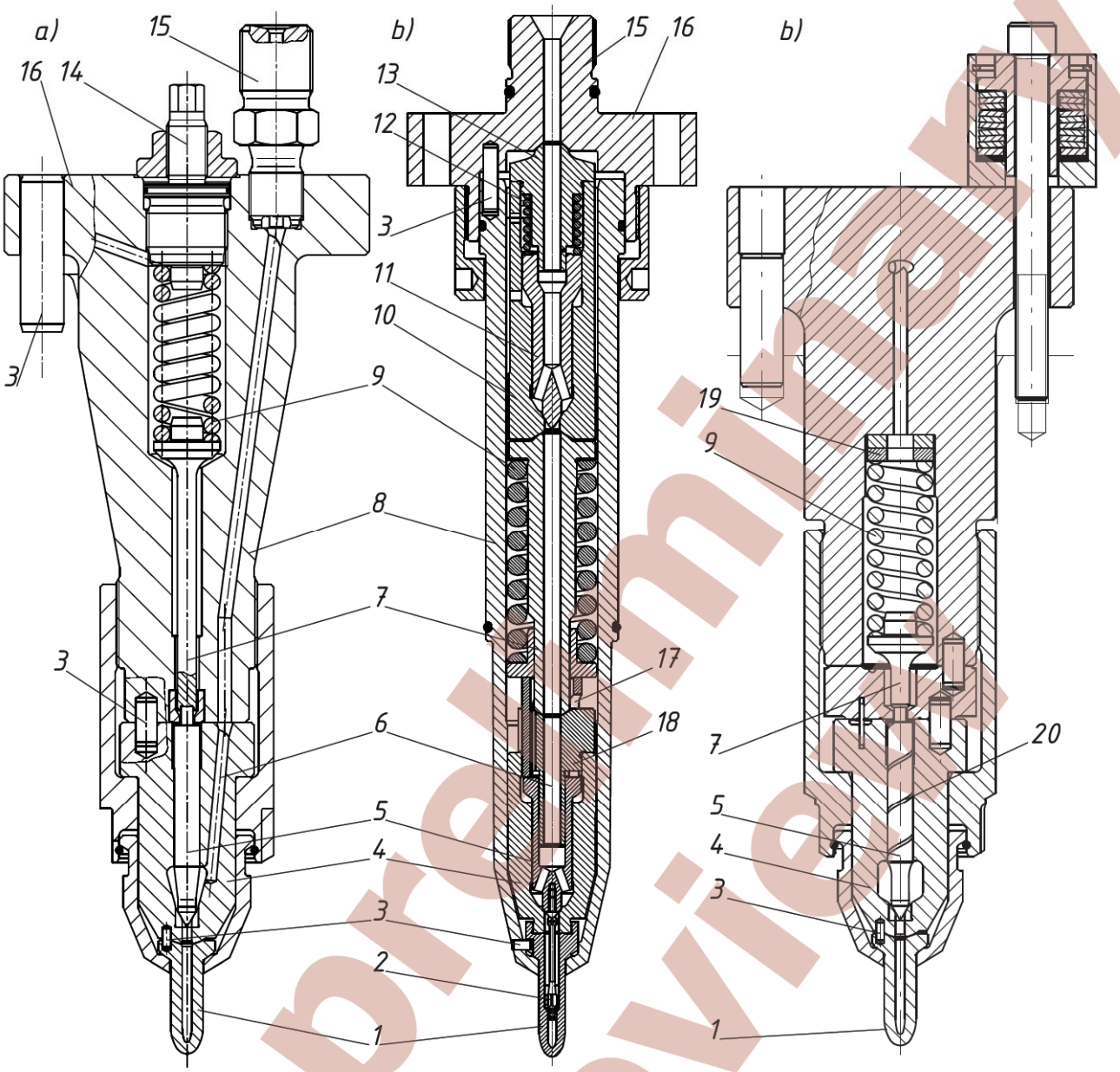


Figure 5.7 – Injectors of marine low-speed diesel engines: *a* – RTA 58 T from Wärtsilä; *b* – MC series from MAN; *c* – RT-flex 50B from Wärtsilä; 1 – nozzle tip with holes; 2 – hanging valve; 3 – mounting pin; 4 – needle valve housing; 5 – needle valve of the nozzle; 6 – channel for supplying fuel to the fuel injector; 7 – needle valve rod; 8 – fuel injector housing; 9 – loading spring; 10 – circulation valve housing; 11 – circulation valve; 12 – circulation valve spring; 13 – circulation valve stop; 14 – screw for adjusting the initial tightening of the loading spring; 15 – fuel supply fitting; 16 – fuel injector mounting flange; 17 – intermediate stop; 18 – needle valve guide; 19 – adjusting washer; 20 – throttling groove (adapted from [16, 17, 18])

The compound nozzle simplifies the production of precision pairs and is less subject to thermal stress. For low-speed diesel injectors, multi-jet nozzles are used, better are more consistent with mixture formation in undivided combustion chambers (Fig. 5.8).

For injectors with a peripheral location, the openings of the nozzle tips should form aerosol jets covering a certain area of the combustion chamber. As a result, the nozzle holes have to be located on one side of the fuel nozzle tip, slightly shifting them by a certain angle and height (Fig. 5.8 *c*). In this case, the nozzle tip has to be lengthened, thereby increasing the area of the protruding part and, consequently, the amount of heat perceived by it. In addition, the volume of the sub-needle sac increases, in which, after injection is completed, the remaining fuel is exposed to high temperatures. At the same time, a certain amount of fuel evaporates and, through the nozzle holes, fuel vapors en-

For preliminary review

ter the working cylinder. Some of the vapor at the afterburning stage manages to partially burn out with the formation of soot, while the part that enters the working cylinder after the combustion process is completed increases the content of hydrocarbons in the exhaust gases. In addition, in the space under the needle valve and on the surface of the fuel nozzle tip, under the influence of high temperatures, the process of fuel pyrolysis occurs with the formation of solid coke, which can disrupt the normal conditions of the injection process.

For the reason given above, all manufacturers of fuel systems for marine low-speed diesel engines are trying to reduce the volume of the under-head chamber. Thus, in its new developments, Wärtsilä used a tip pressed into the housing, aligning it with the needle valve seat (Fig. 5.8 *b*).

MAN has used an additional cone-type suspension valve, which, when the main valve is closed, cuts off part of the sub-orbital chamber, leaving only a volume corresponding to the position of the uppermost fuel injector opening (Fig. 5.8 *d*).

The problem of reducing the sub-needle chamber volume is most fully solved in the design of the MAN company, which is presented in Fig. 5.8 *e*. In this design, the company used a sliding spool located in the channel of the sub-needle chamber, connecting it to the main valve of the nozzle. The disadvantage of this design is that all fuel injector holes must be located at the bottom of the fuel nozzle tips. As a result, the holes have a fairly large diameter, which leads to deterioration in the atomization quality. Therefore, injectors with such a nozzle are used primarily on engines with a hydraulically driven fuel pump, in which the fuel injection pressure is practically independent of the engine speed.

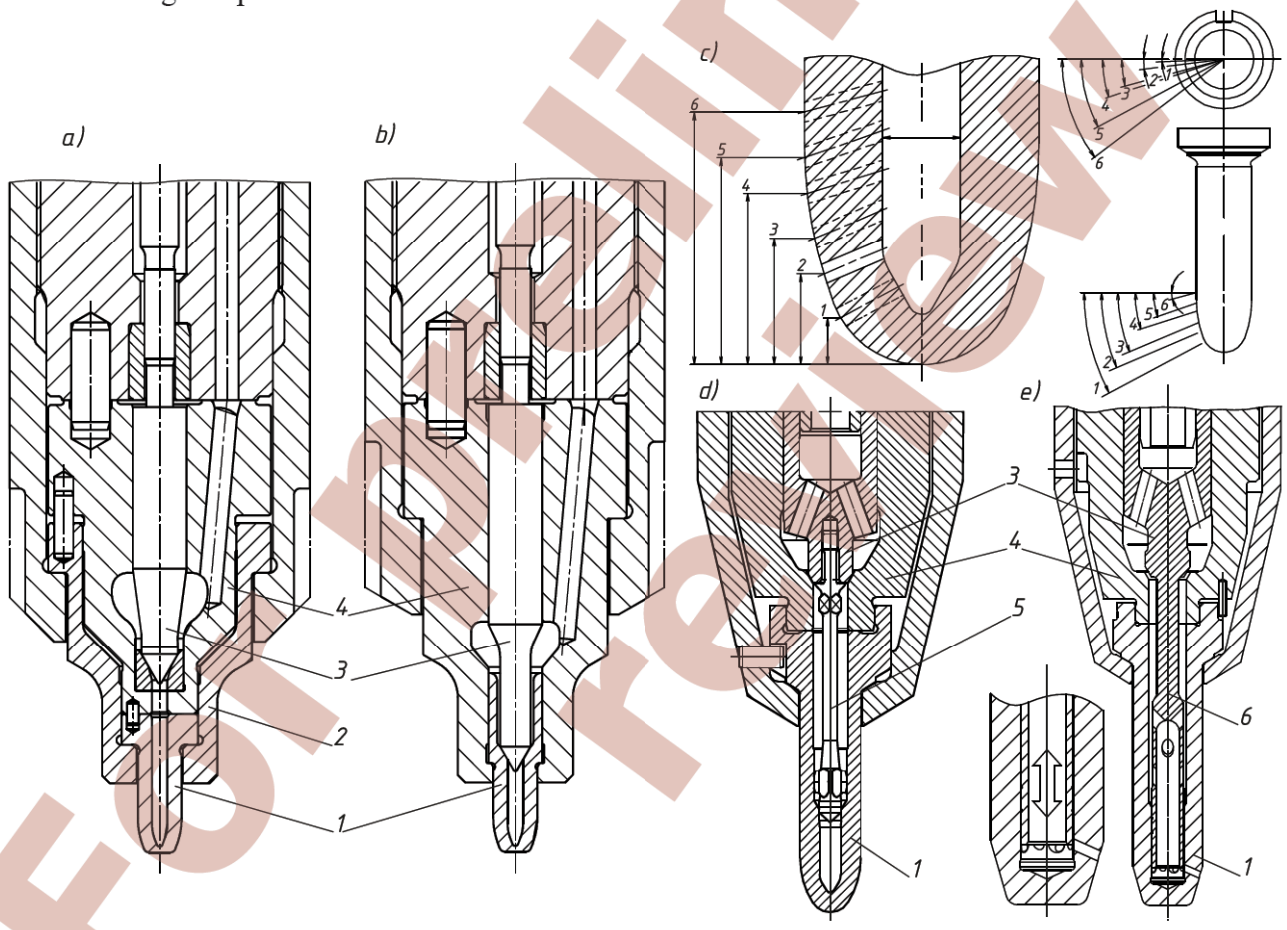


Figure 5.8 – Fuel injectors for low-speed diesel engines. *a* – with a replaceable fuel nozzle tip and an insertable needle valve seat (engines of the RTA series from Wärtsilä); *b* – with a replaceable fuel nozzle tip combined with a needle valve seat (engines of the RTA series from Wärtsilä); *c* – location of holes in the tip of the fuel nozzle tip (MC series engines from MAN); *d* – with an additional suspension-type valve (MC series engines from MAN); *e* – with an additional spool-type valve (MC and ME series engines from MAN). 1 – replaceable fuel nozzle tip; 2 – outer housing of the nozzle; 3 – needle valve; 4 – needle valve housing; 5 – suspended cone valve; 6 – spool valve of the fuel nozzle tip (adapted from [17, 19, 20])

For preliminary review

The main feature of fuel supply in engines equipped with accumulator injection systems is the presence of constant pressure in the accumulator, which limits the possibility of its sudden release in front of the injector. In a accumulator injection systems, the pressure in the space under the needle will drop only due to the flow of fuel through the fuel injector holes. As the pressure in the supraglottic space drops, the needle will lower, thereby increasing the hydraulic resistance in the gap between the shut-off cone and the needle valve seat. As the gap decreases, the rate of fuel flow from the fuel injector openings will decrease and at the end of injection may not be sufficient to flush the remaining fuel from the fuel nozzle tip. The resulting drop will begin to coke under the influence of high temperatures, and gradually the tip will stop working.

If there is a helical groove (Fig. 5.7 *c*), which has a sufficiently long length, the injection process will proceed as follows: when a relatively large portion of fuel is supplied to the injector from the control unit, the hydraulic resistance of the groove is too large to significantly affect the leakage of fuel from the fuel injectors flange space (the throttling effect of the groove is very large). But when the needle lands, when the fuel supply has stopped, at some point the resistance of the groove will become less than the resistance in the gap between the needle valve and its seat. In this case, the fuel from the space above the needle will follow the path of least resistance, that is, through the helical groove into the spring chamber and then to the drain, and the needle valve will quickly close, ensuring a sharp cutoff of injection.

Another important feature of fuel supply in low-speed diesel engines is the need to maintain the thermal regime of all elements of the fuel equipment to ensure the specified viscosity of heavy fuel. This is especially true when the engine is stopped, since a decrease in fuel temperature can lead to an unacceptable increase in viscosity, at which operation of the fuel system will be impossible. In early designs, this problem was solved by switching the engine to low-viscosity fuel before stopping, which filled the fuel supply system and ensured reliable starting of the engine from a cold state. Today, this procedure is performed only when it is necessary to stop not only the engine itself, but also all its systems.

In some fuel systems, to maintain the temperature, local heaters are installed, so-called satellites, which heat the elements of the fuel equipment due to the heat of the water vapor supplied to them.

Currently, fuel systems with constant circulation of heated fuel are widely used. In this case, the fuel circulates not only while the engine is stopped, but also during the periods between injections. This ensures not only maintaining the specified fuel viscosity, but also cooling the injectors.

To ensure constant circulation, a number of changes have been made to the design of the fuel system elements, the main of which are the replacement of the high-pressure fuel injection valve with a filler valve and the installation of circulation valves in the injectors. The heated fuel, supplied by an electrically driven booster pump during the periods between injections, enters the injection pump space above the plunger through the open delivery valve, from which it enters the injector through a high-pressure pipeline. Next, the fuel enters the cooling cavity of the fuel injector through the open circulation valve, from which it is drained back into the supply tank through the drainage channel.

The operating diagram of a MAN low-speed engine fuel injector equipped with a circulation valve is shown in Fig. 5.9.

A needle-type circulation valve installed at the top separates the injector high-pressure cavity from the fuel supply line. In the absence of supply, fuel enters the cavity of the circulation valve with a pressure of about 1.0 MPa created by the booster pump. This pressure is not enough to open the valve, overcoming the force of the spring loading it. There is a small drain hole in the upper valve guide through which fuel flows from the circulation valve cavity into the cooling cavity of the injector. Next, through the drain fitting, the fuel is discharged back into the supply tank (Fig. 5.9 *a*).

At the beginning of the injection stroke of the injection pump, the drainage hole is unable to drain all the fuel coming from the injection line. As a result, the pressure in the valve cavity begins to increase, which leads to its opening. When the valve is lifted, the drainage hole on the guide is blocked, and the valve cavity is disconnected from the drain line. From this moment, all the fuel supplied by the injection pump enters through the open circulation valve into the cavity of the fuel

For preliminary review

injector (Fig. 5.9 *b*).

When the pressure required to open the fuel injector valve is reached, the valve rises, simultaneously moving the fuel nozzle tip spool valve and allowing fuel access to the fuel nozzle tip openings. Injection begins (Fig. 5.9 *c*).

At the end of the injection stroke of the injection pump plunger, all valves return to their original position.

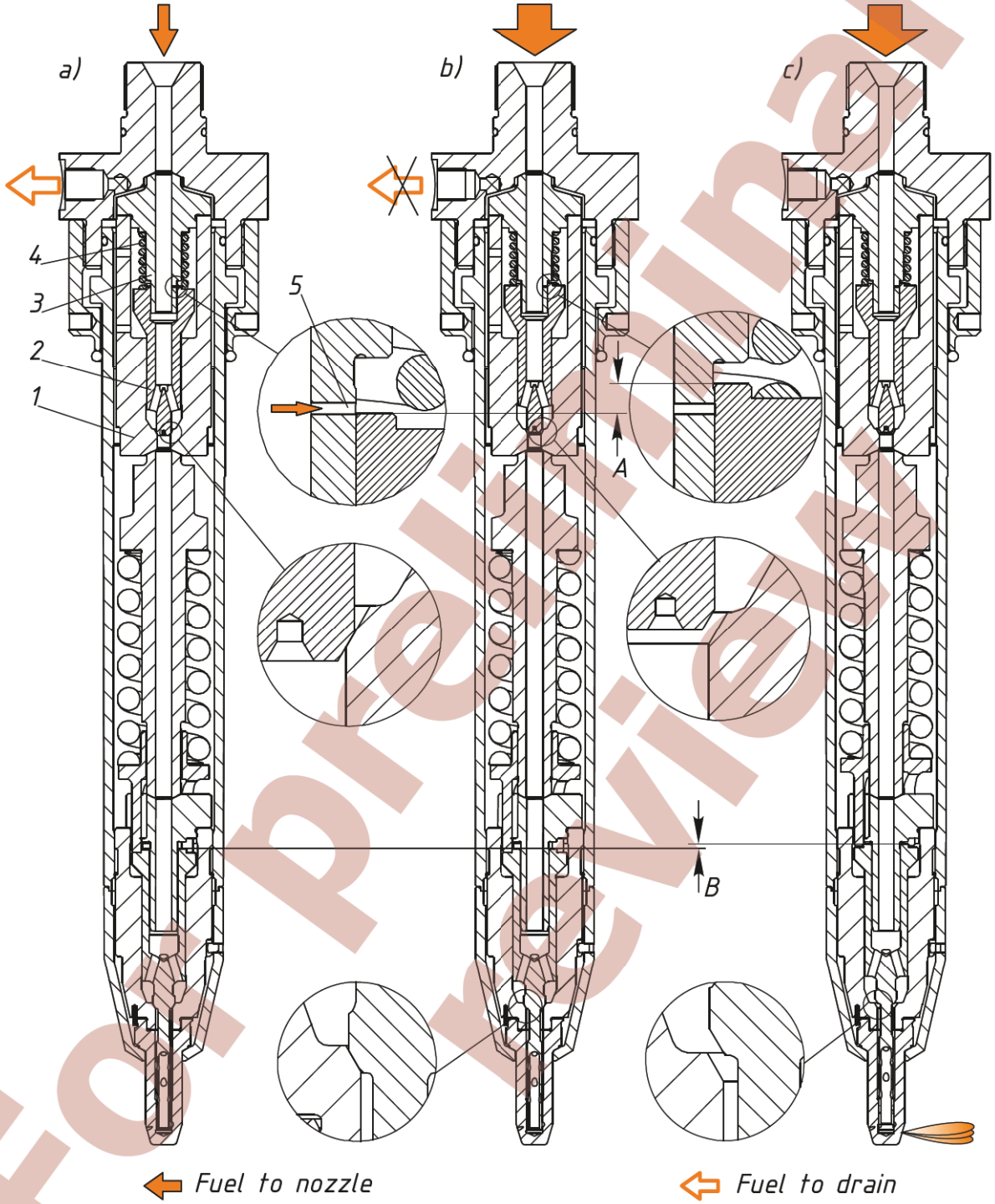


Figure 5.9 – The order of operation of the injector of low-speed diesel engines of the MC and ME series from MAN: *a* – the circulation valve is closed, the fuel goes to the drain through the drainage hole; *b* – the pressure in the cavity of the circulation valve increases, the valve opens, rising upward, and closes the circulation hole; *c* – the circulation valve is open, the pressure in its cavity increases, the valve rises, allowing fuel access into the fuel injector cavity, and the spool rigidly connected to the main valve opens the fuel injector holes. 1 – circulation valve seat; 2 – circulation valve; 3 – valve guide; 4 – spring; 5 – drainage hole (adapted from [20])

For preliminary review

In Fig. 1.10 shows the design of the circulation valve used by Wärtsilä for injectors of low-speed engines of the RTA series. The valve is located in the fuel injector connecting fitting. According to the principle of operation, it is similar to the previously discussed valve from MAN. The only difference is that for a more reliable separation of the valve cavity from the drain line, in addition to the spool pair, an additional seal is provided here by installing a check valve consisting of a shut-off cone, which sits on a seat in the guide sleeve. The fuel enters the external cavity of the valve through the drain hole, from where it is diverted through a special channel to cool the nozzle, and then enters the drain fitting installed on the upper flange of the fuel injector.

A significant increase in the maximum cycle pressure in modern low-speed engines has led to the need to increase the opening pressure of the nozzle needle valve. This was dictated by two main considerations. Firstly, an attempt to improve the quality of fuel atomization at the initial stage of the injection process, and secondly, to prevent the reflux of gases from the combustion chamber into the fuel system.

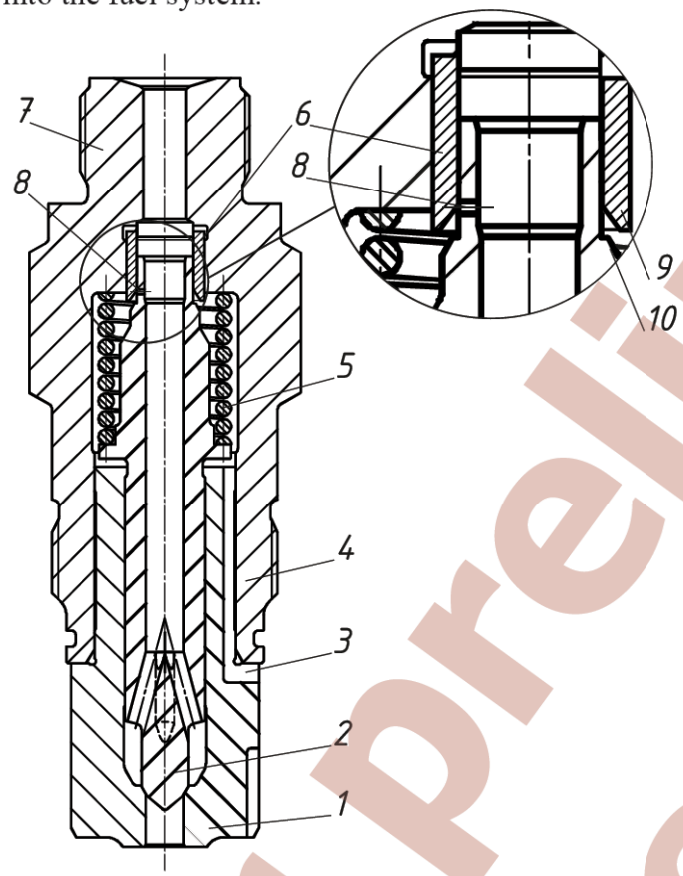


Figure 5.10 – Circulation valve of Wärtsilä RTA series engines: 1 – valve seat; 2 – needle valve; 3 – channel for draining fuel; 4 – valve housing; 5 – spring; 6 – upper guide of the needle valve with a locking seat; 7 – connecting fitting; 8 – drainage hole; 9 – check valve seat; 10 – check valve shut-off cone (adapted from [21])

However, for engines operating directly on the propeller, in which power is controlled by changing the speed, such an increase can lead to unstable and uneven supply at low speed modes. Deterioration of combustion conditions leads to increased carbon formation in the combustion chamber and the formation of deposits in the sub-piston cavities, which worsen the gas exchange conditions of the engine.

The performance of engines at low loads can be significantly improved by reducing the pressure at which the fuel injector starts supplying. However, when switching to modes close to maximum power, the pressure must be increased.

Previously, on marine engines, the problem of regulating the opening pressure of the spray valve was solved by organizing additional hydraulic loading of the needle valve. For this purpose, the engine was equipped with a special system for supplying fuel or oil to the injector hydraulic locking system. By changing the pressure in this system, you can change the pressure at the start of fuel injection by the injector. Currently, such a system on marine diesel engines is used only in injectors with electrohydraulic control of the accumulator-type injection process (the so-called Common rail systems).

The design and principles of operation of such fuel injectors will be discussed further in the rele-

For preliminary review

vant chapters.

In Fig. 5.11 shows an injector design developed by MAN, which contains a device for changing the pressure at which the injector starts to open depending on the average fuel injection pressure.

The device consists of a fuel supply channel to the nozzle and two cylindrical pistons, with the help of which the preload of the main needle valve spring is changed. Initially, the spring is adjusted to open the valve at a pressure of about 20 MPa (graph in Fig. 5.11).

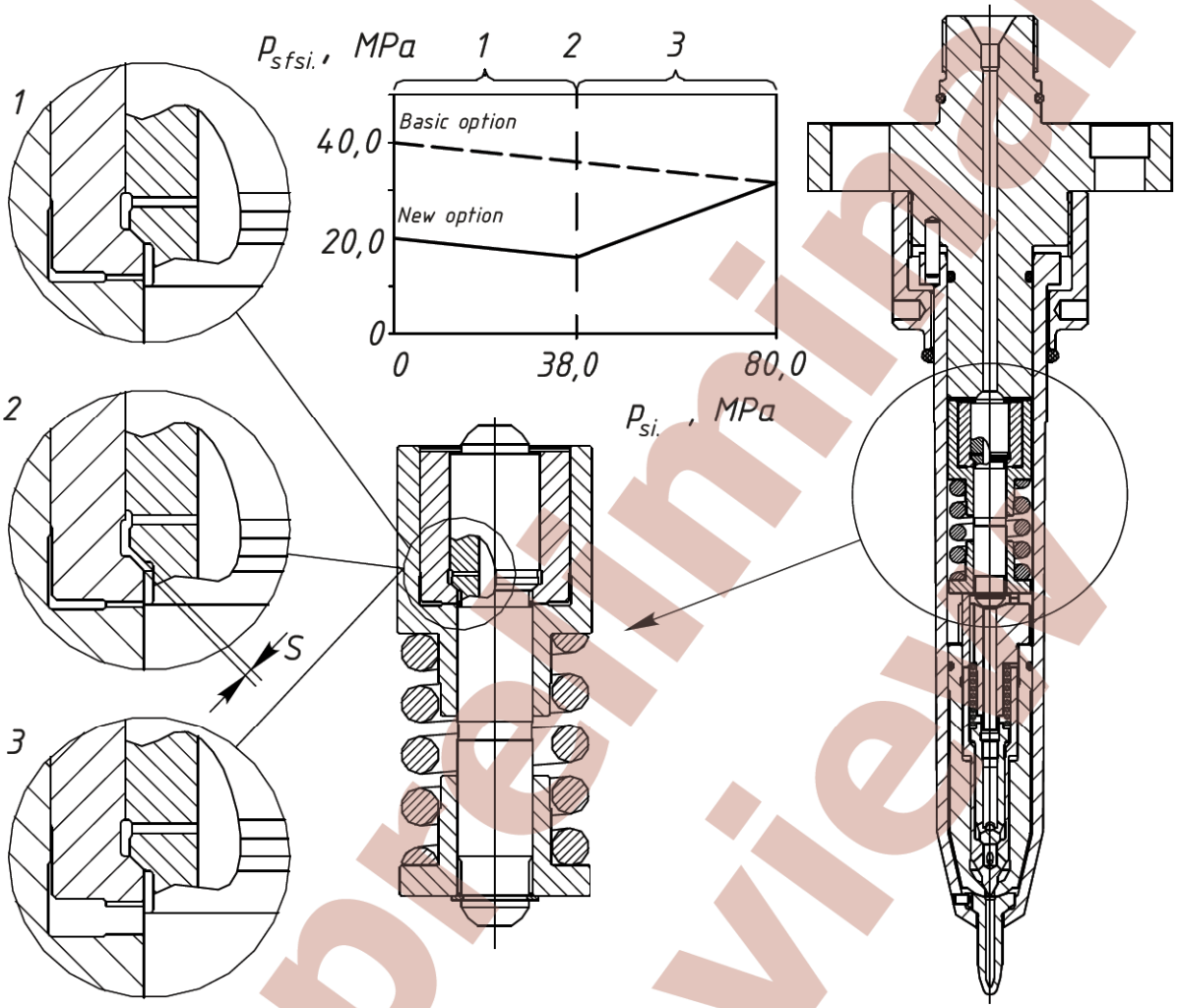


Figure 5.11 – Injector of a low-speed engine with a mechanism for automatically changing the injection start pressure and a lower location of the circulation valve: *a* – drain hole of the circulation valve; *b* – drainage hole of the needle valve spring tightening control valve; 1 – needle valve spring; 2 – external piston; 3 – intermediate piston; 4 – fuel supply channel to the needle valve (adapted from [22])

When the injector operates, part of the fuel flows through the drainage hole from the fuel supply channel into the cavity formed by the outer wall of the channel and the intermediate piston. The outer protrusion on the outer surface of the channel and the inner protrusion on the intermediate piston, made at an angle of 45°, form a valve pair. As a result of fuel throttling, a certain constant pressure is established in the cavity above the valve pair, the value of which depends on the injection pressure. If the pressure in this cavity is not sufficient to open the valve, the outer piston with its protrusion rests against the end of the intermediate piston, keeping the pre-tension of the spring unchanged. As the injection pressure increases, the increase in pressure in the cavity of the valve pair leads to its opening (gap *S* in Fig. 5.11) and part of the fuel flows into the cavity between the end of the intermediate and the protrusion of the outer cylinder, forcing the outer piston to move down, thereby increasing the initial tightening of the needle valve spring. As a result, the pressure at the beginning of the opening of the needle valve of the nozzle increases in proportion to the increase in injection pressure, up to 38 MPa.

For preliminary review

Since two-stroke engine injectors contain a large number of moving parts, many of which are precisely fitted to each other, it is extremely important that when installing the injector into the cylinder cover, no stress is created that could lead to their deformation. For this, special shock absorbers are used, which are a cylinder filled with a set of disc springs (Fig. 5.12).

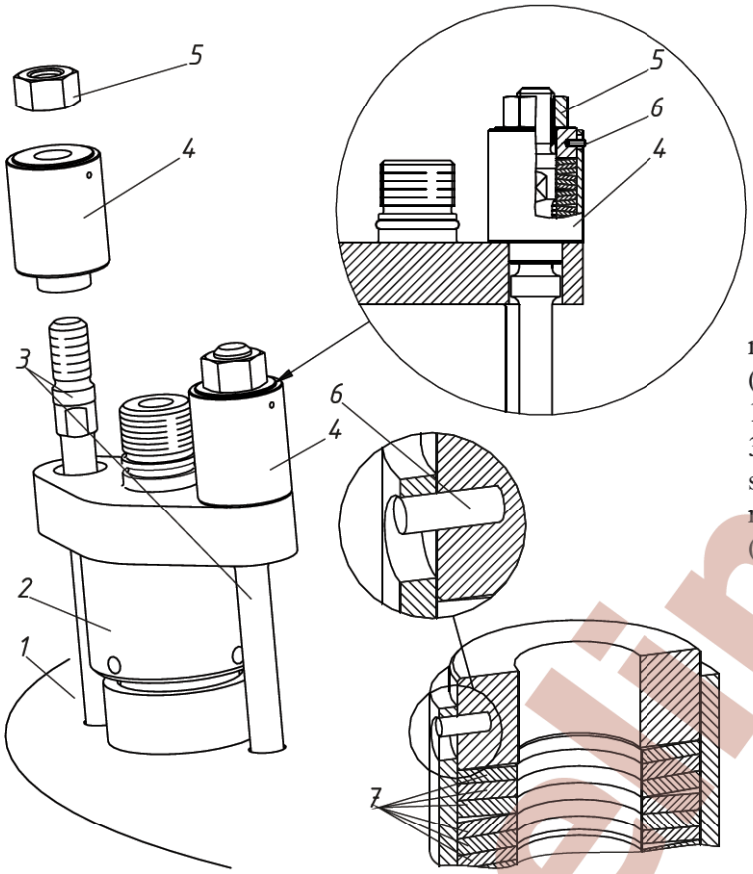


Figure 5.12 – Elastic shock absorber for normalizing the injector tightening value (MC and ME series engines from MAN): 1 – cylinder cover; 2 – fuel injector; 3 – fuel injector mounting studs; 4 – elastic shock absorber; 5 – fuel injector fastening nut; 6 – control pin; 7 – set of disc springs (adapted from [17])

The shock absorbers are put on the mounting studs and rest their bottom on the fuel injector flange. The tightening force from the nuts through the upper washer is transmitted to the fuel injector flange through a set of springs.

The shock absorber housing has a control hole in which a pin is placed, pressed into the upper washer. When properly tightened, the pin occupies a central position in the control hole.

High pressure fuel lines serve to supply fuel from the injection pump to the injectors. To reduce losses in supply pipelines, they try to make them as short as possible. Currently, two types of fuel pipelines, shown in Fig. 5.13.

In the first case, a steel tube is installed between the injection pump and the fuel injector fitting, covered with a protective shell on top (Fig. 5.13 *a*). The casing serves to prevent fuel leakage in the event of damage to the main pipeline. In addition, the space between the high pressure pipe and the protective casing is used to collect and drain leaks from the fuel injector and the connections of the pipeline itself. The ends of the tubes are made in the form of conical thickenings, which are pressed against the wells of the fittings using union nuts. The fuel supply fitting to the injector itself is pressed against the fuel intake well of the injector using an elastic clip. This allows you to avoid excessive pressure in the pipeline in case of fuel injector jamming. The fitting, under the influence of pressure, will overcome the pressing force of the clip, and the fuel supplied by the injection pump will drain into the drainage channel.

The pipeline design shown in Fig. 5.13 *b*, consists of two fittings passing inside the drillings in the cylinder cover. The fittings are pressed against the wells of the fuel receiving channel of the injector and the mating surface of the fuel pump, as well as against each other, using special screw plugs that are screwed into the lug on the cylinder cover. One of the plugs contains an elastic element that protects the high-pressure line from excessive loads. The internal cavity of the tide together with the protective casing forms a leakage collection box. The cavity formed between the

For preliminary review

drilling in the cylinder cover and the fuel supply fitting to the injector is used to drain leaks.

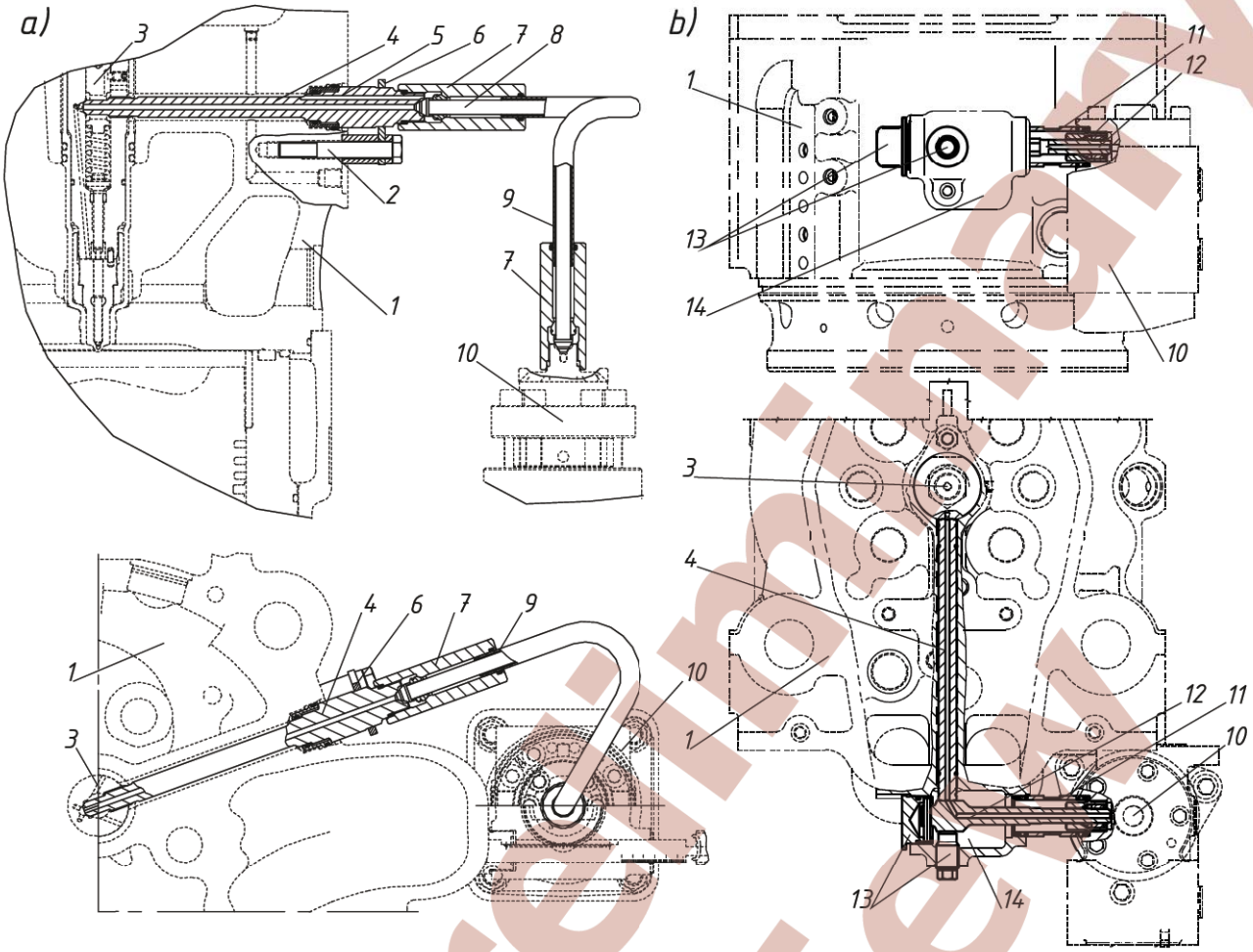


Figure 5.13 – High-pressure lines of medium- and high-speed diesel engines: *a* – high-speed diesel L27/38 from MAN; *b* – medium-speed diesel engine M 32 C from MaK; 1 – cylinder cover; 2 – bolt securing the fuel fitting; 3 – fuel injector; 4 – fuel fitting; 5 – leakage drainage channel; 6 – spring clip for fastening the fuel fitting; 7 – union nut for fastening the high-pressure pipeline; 8 – high pressure pipeline; 9 – outer shell of the high-pressure pipeline; 10 – injection pump; 11 – protective casing; 12 – fuel fitting; 13 – thrust plugs; 14 – box for collecting leaks (adapted from [23, 24])

Two-stroke diesel engines can have two or three injectors per cylinder. The number of high pressure lines corresponds to the number of injectors. In Fig. 5.14 shows a general view and location of the high pressure lines on the cylinder cover. Wärtsilä RTA series engines have a mid-mounted fuel pump, from which fuel is supplied through a common pipe to the fuel distributor and then from there through separate pipelines to the injectors (5.14 *a*). The fuel pipes of the distant injectors are made detachable for ease of installation. And the presence of a massive flange connection on the tubes reduces their resonant frequency. All high-pressure pipes are placed in corrugated sleeves made of rolled steel tape. These sleeves are covered on top with a sheath of braided steel wire. The space between the pipe and the protective casing communicates with the cavity of the leakage collection box through drillings in the flanges. The outer shell also functions as a heat shield, protecting the high-pressure pipeline from rapid cooling.

The high-pressure lines of MAN MC series engines are designed in a similar way. The main difference is that the use of a camshaft chain drive allows high-pressure pumps to be placed in close proximity to the cylinder covers. Therefore, these engines do not use intermediate pipelines and dividers, and the fuel flows directly from the pump cover into the pipeline of the corresponding fuel injector.

For preliminary review

As with the RTA series engines, the spaces between the pipe and the protective casing are used to collect leaks, which are connected through drillings in the flanges to drain holes in the pump cover.

Each fuel pump, in turn, is connected via an outlet tube to a common drain tank in which a level switch is installed. The drain tank is equipped with a bypass valve, in the lower part of which there is a drain tube, from where small leaks are discharged outside without affecting the level switch.

In the event of cracks in the pipelines or large leaks in the system, the flow area of the said tube is not enough to drain the increased amount of fuel, and the fuel level in the drain tank becomes sufficient to trigger the bypass valve. An increased fuel level causes a signal to be sent from the level relay.

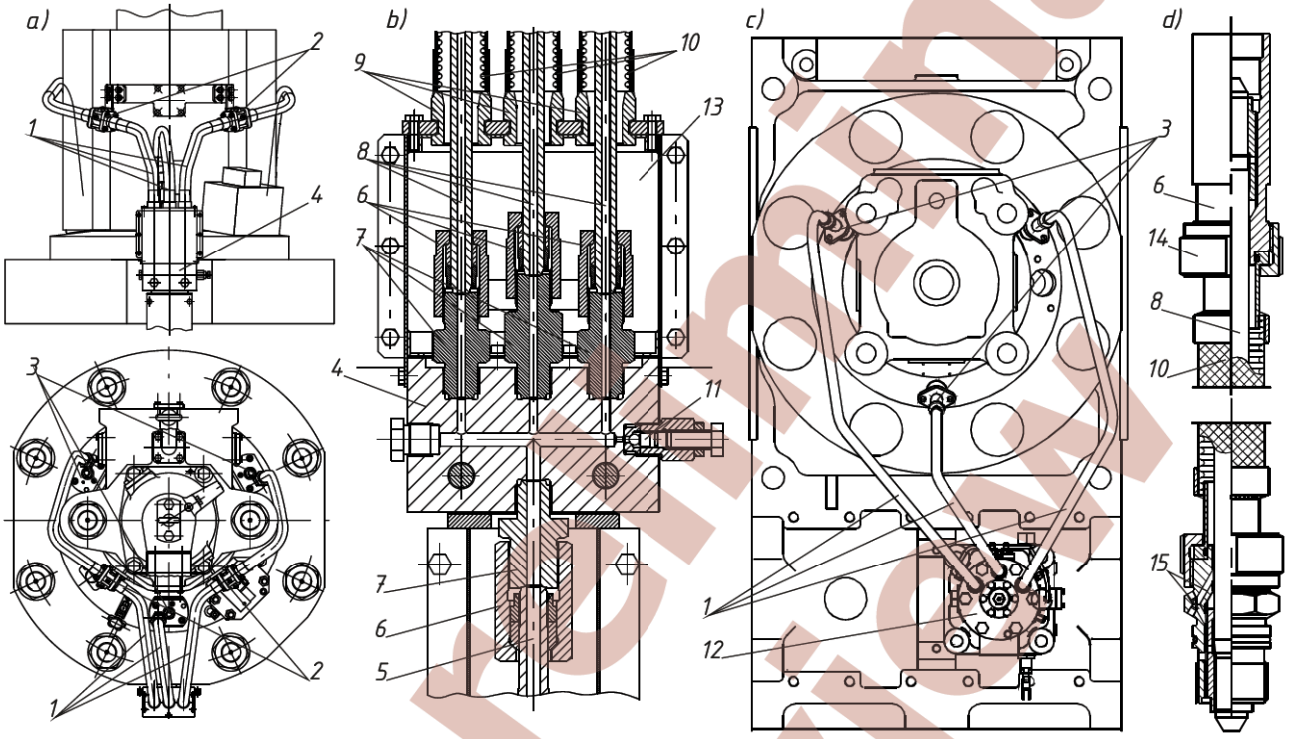


Figure 5.14 – High-pressure lines of low-speed diesel engines: *a* – location of high-pressure pipelines on the cylinder cover of the RTA 72 U diesel engine from Wärtsilä; *b* – fuel distribution device for diesel injectors RTA 72 U from Wärtsilä; *c* – location of high pressure pipelines on the K98MC engine from MAN; *d* – design of the pipeline and connecting devices of the K98MC engine from MAN; 1 – high pressure pipelines; 2 – connecting devices; 3 – fuel injectors; 4 – device for distributing fuel to injectors; 5 – high pressure supply line; 6 – union nut for fastening high-pressure pipelines; 7 – connecting fitting; 8 – internal high pressure pipeline; 9 – fitting for fastening the outer shell of the high-pressure pipeline; 10 – outer shell; 11 – valve for pumping the fuel distributor; 12 – injection pump; 13 – leakage collection box; 14 – on the union nut for fastening the outer shell; 15 – leakage collection channel (adapted from [25, 26])

5.4.3 Spool-controlled fuel pumps

Due to their simplicity and relatively low cost, pumps with spool-type supply control are most widely used on marine diesel engines of all types. The main element of the pump is a plunger pair, consisting of a bushing and a plunger precision fitted to it. The gap between these parts, depending on the size of the pair and the viscosity of the fuel used, lies within 5...12 μm. Such precise adjustment allows to minimize inevitable leaks in the plunger pair. To regulate the active stroke of the plunger, an inclined groove is cut on its housing, acting as a spool, and the housing of the sleeve has one or more holes for filling the space above the plunger and draining excess fuel. During the injection stroke, the plunger is driven by the fuel cam, and the reverse stroke occurs due to the compression force of the return spring.

The principle of spool control of cyclic supply is shown in Fig. 5.15.

For preview only

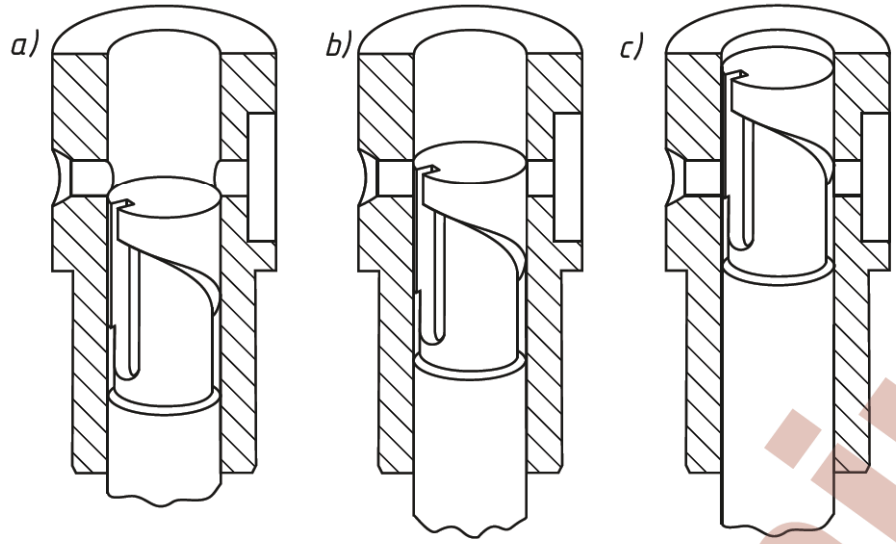


Figure 5.15 – Stages of the fuel supply process by a section with spool control: *a* – filling the space above the plunger; *b* – start of injection; *c* – injection end (adapted from [9])

When the plunger is in the lower position, it completely opens the filling hole, through which fuel supplied from the booster pump under a pressure of 0.1...0.5 MPa fills the space above the plunger (Fig. 5.15 *a*). As the plunger rises, its upper edge gradually covers the filling hole and, when it is completely closed, the discharge (active) stroke of the plunger begins (Fig. 5.15 *b*). The supply of fuel to the high-pressure line and further to the fuel injector continues until the oblique edge, the cavity of which communicates with the space above the plunger, coincides with the filling hole (Fig. 5.15 *c*). In this case, when the plunger moves upward, the fuel is not pumped, but simply flows from the space above the plunger into the low-pressure cavity. The part of the plunger stroke during which fuel is not pumped is called the idle stroke.

In addition to translational movement, the plunger of a spool-controlled fuel pump can rotate at a certain angle relative to its axis. In this case, the position of the inclined edge relative to the hole in the bushing changes, as a result of which the magnitude of the active stroke, and, consequently, the cyclic portion of fuel decreases (Fig. 5.16).

To rotate the plunger relative to its axis in marine diesel engines, a pair is used – a gear rack, a gear sector (Fig. 5.17). In individual pumps, the rack rotates only one plunger; in block pumps, one rack is used to simultaneously drive all pump sections.

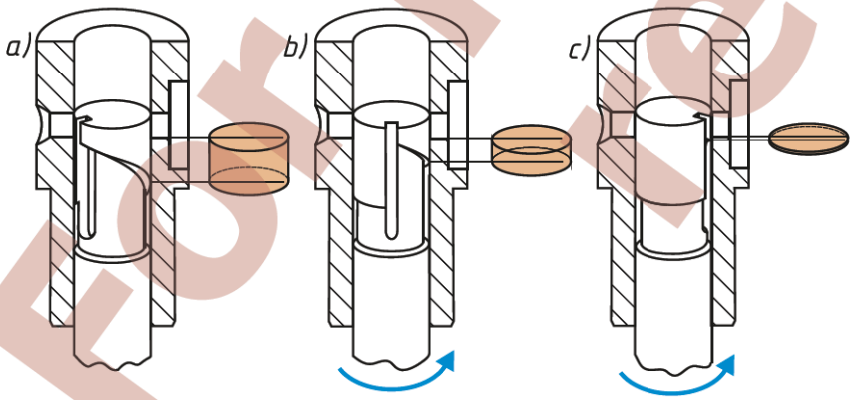


Figure 5.16 – Change in the cyclic portion of fuel when turning the plunger relative to its axis: *a* – full supply; *b* – partial supply; *c* – zero supply (adapted from [9])

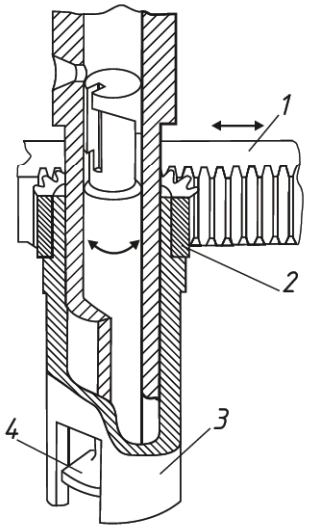


Figure 5.17 – Rack and pinion gear for turning the plunger: 1 – toothed rack; 2 – gear sector; 3 – control sleeve; 4 – plunger shank (adapted from [9])

For preliminary review

The rack is connected through a rod system to the speed controller directly or through a hydraulic booster system.

In spool pumps, three main methods are used to regulate the cyclic portion and phases of fuel supply to the working cylinders (Fig. 5.18).

With the first control method (Fig. 5.18 *a*), the start of fuel supply (point *A*) remains unchanged regardless of the engine speed and load. End of injection (points B_1, B_2 and B_3) changes due to a change in the position of the cut-off edge when the plunger is rotated relative to the bypass holes. This method of regulation is called *end-of-supply regulation*.

In pumps with this type of regulation, the end of the flow at all loads occurs in the area where the plunger rises, where its speed is close to maximum. This ensures high pressure and injection speed throughout the entire fuel supply section, resulting in good fuel atomization quality.

With the second control method (Fig. 5.18 *b*), the beginning of the fuel supply changes (points A_1, A_2 and A_3), and the end of the supply remains unchanged (point *B*). In these pumps the bevel edge is located at the top plunger when moving upward, the edge, depending on the rotation of the plunger, sooner or later closes the inlet hole in the sleeve, after which its active stroke begins. The end of the active stroke corresponds to the beginning of the cutoff (point *B*). Thus, in a pump of this type, with a change in the cyclic supply, the fuel injection advance angle simultaneously changes. This method of regulation is called *regulation at the beginning of supply*.

Rotating the plunger leads not only to a reduction in the cyclic portion of fuel, but also to a decrease in the supply advance angle. This leads to the fact that the beginning of the supply is shifted to a section of higher speeds of movement of the plunger (from point A_1 to point A_3), which leads to a more intense increase in pressure at the initial stage of injection and helps to improve the quality of atomization and combustion of fuel. This is especially important when the engine is running at low loads, since as the speed decreases, the injection pressure drops proportionally. Reducing the advance angle and intensifying injection makes it possible to optimize the heat release process while reducing the rotation speed and load on the engine.

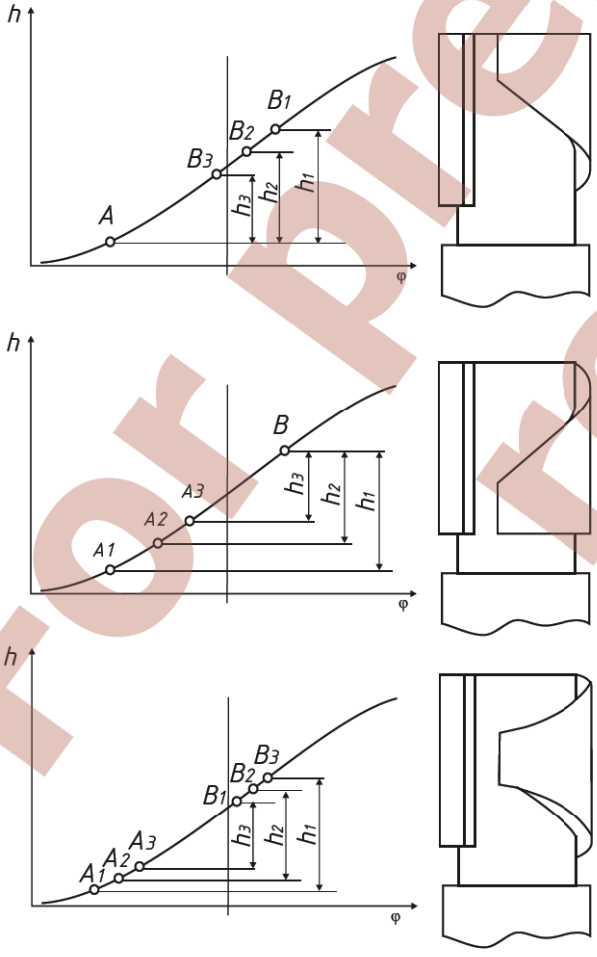


Figure 5.18 – Methods for regulating the cyclic portion and timing of fuel supply to the engine cylinders: *a* – at the end of supply; *b* – at the beginning of the supply; *c* – at the beginning and end of the supply (adapted from [9])

For preliminary review

With the third control method, the plungers have two oblique edges that control the beginning (points A_1, A_2, A_3) and the end (points B_1, B_2 and B_3) of the supply (Fig. 5.18 *c*).

When the engine load changes, turning the plunger leads not only to a change in the amount of fuel injected, but also to changes in both the beginning and end of the supply. This method of regulation is called *regulation at the beginning and end of the supply*. It combines the advantages of the two methods discussed above and at the same time is free from their disadvantages.

In a number of its designs, MAN uses mixed spool control, which combines control methods at the end and beginning of the supply (Fig. 5.19).

At large cyclic supplys (section 0-1), only the end of the supply is adjusted, while the upper edge of the plunger is lowered relative to its end by a certain amount, which makes it possible to shift the start of injection to a higher-speed section of the plunger lift. When switching to a reduced load mode (approximately 85% of N_e), the plunger rotates relative to the inlet opening with section 1-2, where the upper edge rises. Thanks to this, the hole in the bushing is closed by the plunger earlier and the advance angle is reduced, which allows maintaining maximum pressure when the load is reduced.

In section 2-3, the supply start angle remains unchanged, and the cyclic portion is regulated at the end of the supply. When switching to medium and low load modes, regulation is carried out at the beginning and end of the supply (section 3-4).

This geometry of the working edges makes it possible to maintain the maximum cycle pressure p_z in the load range from 85 to 100% at an almost constant level. As a result, diesel efficiency in these modes increases. For comparison, in Fig. 1.19 shows graphs of changes in pressure and specific fuel consumption for injection pumps with regulation at the end of the supply and for mixed regulation.

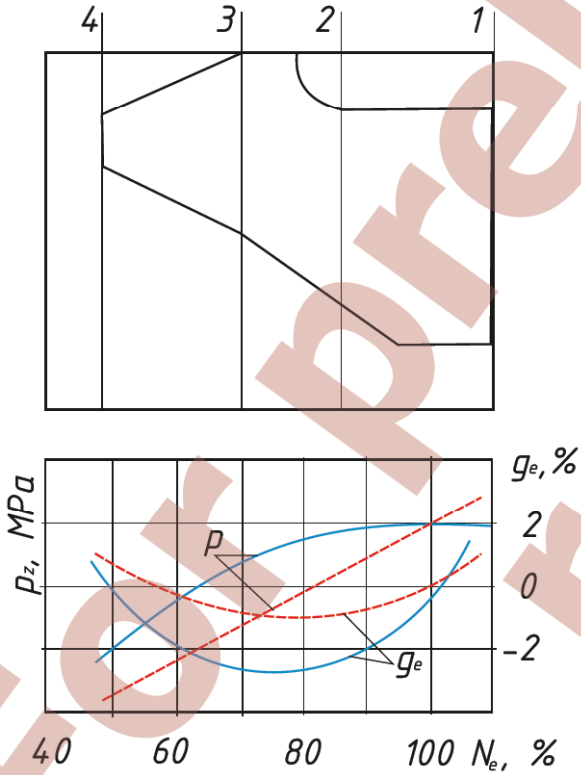


Figure 5.19 – Development of spool edges on a plunger with mixed regulation, as well as graphs of changes in pressure and specific fuel consumption with regulation at the end of supply (- – - -) and with mixed regulation (———) (adapted from [9])

The designs of some pumps with spool control of four-stroke MSE and HSE are shown in Fig. 1.20.

Injection pump valve, like the bushing with the plunger, they are precision parts. In ship pumps, automatic delivery valves are used to separate the high-pressure pipeline and the space above the plunger (Fig. 5.21). The installation of these valves improves the filling of the space above the plunger, maintains a specified residual pressure in the discharge pipeline during the period between injections, and corrects the fuel supply law.

For preliminary review

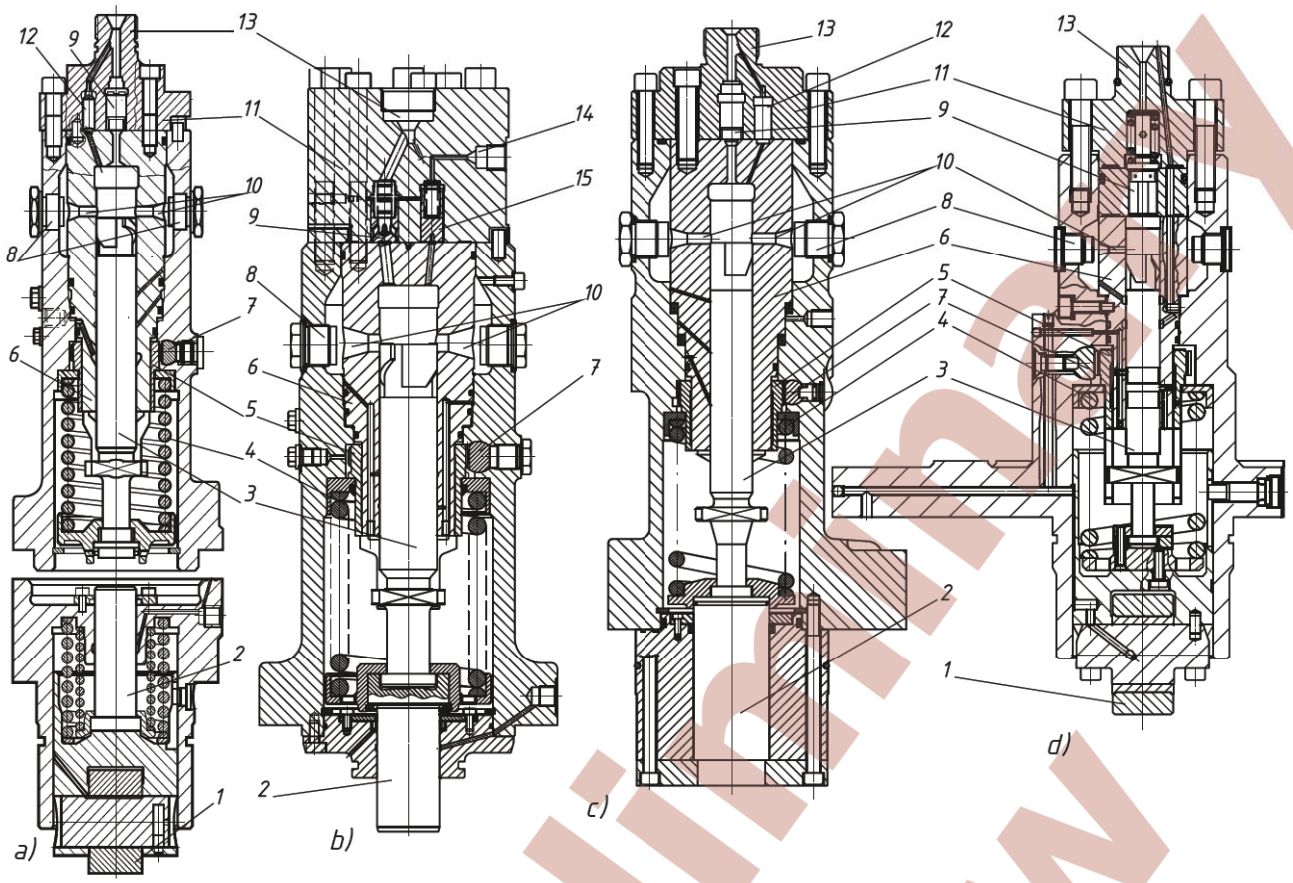


Figure 5.20 – High-pressure fuel pumps with spool valve control of engine cyclic supply: *a* – TBD 645 from Deutz MWM; *b* – Wärtsilä 46 series; *c* – MaK 32C; *d* – MAN L16/24; 1 – roller pusher; 2 – pusher rod; 3 – plunger; 4 – return spring; 5 – plunger reversal sleeve with a gear sector; 6 – plunger bushing; 7 – toothed rack; 8 – anti-cavitation damping devices; 9 – delivery valve; 10 – filling (drain) ports; 11 – pump cover; 12 – reversible valve; 13 – connecting fitting (hole) of the high pressure pipeline; 14 – connecting hole of the high pressure pipeline of the auxiliary fuel injector; 15 – delivery valve of the high pressure line of the auxiliary fuel injector (adapted from [11, 14, 24, 25])

In Fig. 5.21 presents design options for injection valves.

On high-speed diesel engines, *mushroom valves are the most common* (Fig. 5.21 *a*). This type of valve is installed in a removable housing with a seat and consists of three main elements: a shut-off cone, a relief collar and a guide.

Under the influence of fuel pressure, the valve moves along its axis, compressing the spring. To eliminate radial runout when the relief collar enters the hole of the valve housing and when the shut-off cone is seated on the seat, a guide is used, made in the form of a shank with milled grooves for the passage of fuel. The guide surfaces of the shank and the hole in the valve housing are made with high precision. The lower end surface of the valve housing is also precision-engineered. As a rule, it is pressed against the plunger bushing by a fitting, forming a gasketless, hermetic connection with the upper end of the bushing.

At the moment the valve closes, stopping the fuel column leads to a hydraulic shock with the formation of pressure waves, which, in turn, can cause repeated openings of the injector. This phenomenon is called *after-injection*.

The occurrence of after-injection is extremely undesirable, since fuel is reinjected at the expansion line when the main stages of the combustion process have already completed. In addition, fuel is injected into the cylinder under relatively low pressure, resulting in very poor atomization quality. The fuel that gets into the cylinder as a result of injection either does not burn at all, or only manages to burn out partially.

To reduce the likelihood of under-injections, a relief collar is placed on the valve, which fits into the seat when closed. From this moment, the above-plunger cavity and the discharge pipeline are

For preliminary review

disconnected, since the unloading collar forms a gap of 8...15 μm with the channel. Further lowering of the valve leads to an increase in the volume of the cavities forming the high-pressure line, as a result of which the pressure sharply decreases and so-called *unloading occurs*. After the valve is seated on the seat, the above-plunger cavity is completely disconnected from the high-pressure line, in which some excess pressure remains, which makes it possible to reduce the time for its build-up in the next cycle. Thus, the presence of a delivery valve ensures the optimal value of the residual pressure and eliminates injection.

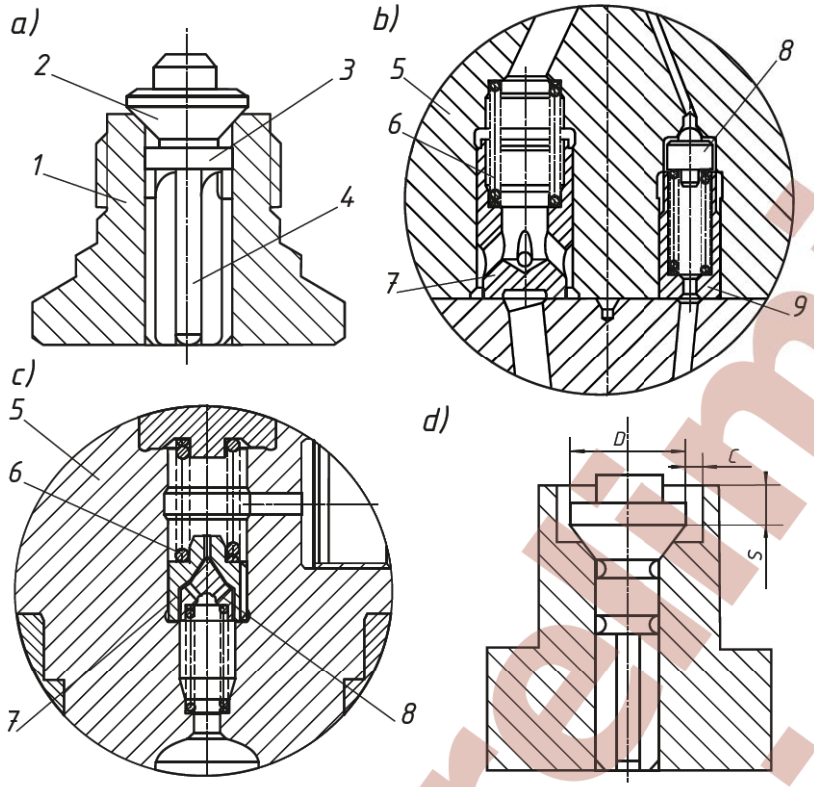


Figure 5.21 – Automatic injection pump valves for marine diesel engines: *a* – conical with an unloading shoulder; *b* – cylindrical with a separately installed check valve; *c* – with a combined discharge and check valve; *d* – corrector valve with hydraulic braking; 1 – valve housing; 2 – shut-off cone; 3 – unloading shoulder; 4 – guide; 5 – fuel injection pump cover; 6 – pressure spring; 7 – cylindrical valve; 8 – check valve; 9 – check valve housing (adapted from [9, 11])

Cylindrical valves (Fig. 5.21 *b*, *c*) are as reliable as fungal ones, but are more compact, therefore they are used on low- and medium-speed diesel engines.

Such a valve does not resist injections well, so *a reversing valve is installed parallel to it,* directed in the opposite direction. Reversing valves reduce hydraulic shock by transferring fuel into the cavity above the plunger and provide stable initial pressure. If the dimensions of the fuel injection pump cover allow, the reversing valve can be made as a separate element (Fig. 5.21 *b*), and if the dimensions are limited, it can be made as one piece with the delivery valve (Fig. 5.21 *c*).

Discharge *corrector valves* allow you to adjust the fuel supply law.

There are many design solutions for such valves, but they all have an unloading shoulder or throttle channels for fuel flow, which interfere with the unloading effect. The corrector valve shown in Fig. 5.21 *d*, has a small recess in the upper part of the housing, into which the locking cone (with a diameter at the base *D*) enters with a gap *C*. In the final stage, the valve seating (section *S*) occurs with hydraulic braking due to the squeezing of fuel from the recess in the housing through the gap *C*. As a result, the unloading effect of the collar is weakened, and the amount of leakage depends on the time of lowering the valve. Therefore, with a decrease in rotation speed, the unloading of the pipeline decreases, and the residual pressure increases. As a result, the injection pressure increases at low speeds due to an increase in the residual pressure in the discharge pipeline.

5.4.4 Spool-type fuel pumps with supply advance angle control

The principle of simultaneous regulation of the cyclic portion and the angle at which fuel supply begins, discussed above, has one significant drawback – the regulation is carried out according to a

For preliminary review

rigid algorithm, which cannot be changed during operation. The need for more flexible regulation may arise when there are significant deviations from the nominal operating modes, when switching to other types of fuel and their mixtures, as well as in a number of other cases. In order to more flexibly regulate the fuel supply process, Burmeister & Wain has developed a spool pump design with a variable fuel injection advance angle (Variable injection timing system hereinafter – VIT). Subsequently, this system was inherited by MAN, which still uses it in its developments to this day.

A general view of the fuel pump with the VIT system, its design and main elements are shown in Fig. 5.22 *a*, *b*, *c*.

In pumps of this type, the advance angle is changed by axial movement of the plunger sleeve relative to the plunger itself. In this case, the position of the cut-off holes relative to the upper edge of the plunger and, consequently, the moment of their overlap relative to the angle of rotation of the engine crankshaft changes.

To use heavy fuels in all engine operating modes, the pumps and injectors are designed in such a way that during parking and during periods between injections, heated fuel circulates in the system, ensuring its heating.

The fuel pump housing has a square base that secures it to the pusher housing. To collect leaks, a special groove is made at the base of the pump, from where leaking fuel flows into a special drain pipe.

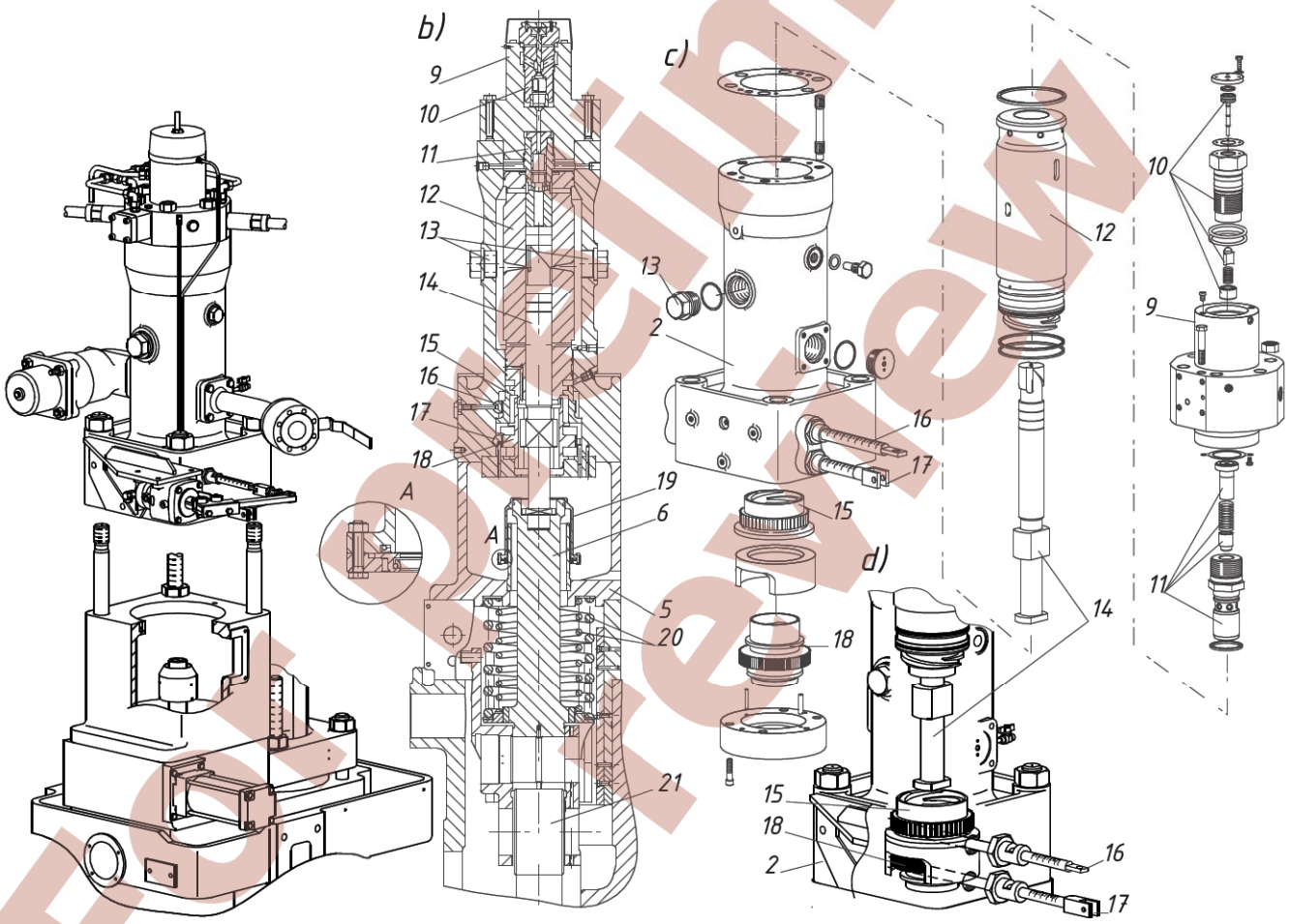


Figure 5.22 – Fuel pump of a low-speed engine of the MC series from MAN with spool control of cyclic supply and fuel injection advance angle (VIT system): 1 – spring pressure damper; 2 – fuel injection pump housing; 3 – high pressure tubes; 4 – fuel supply pipe to the pump; 5 – pusher housing; 6 – pusher; 7 – pneumatic cylinder for driving the reversing mechanism; 8 – camshaft housing; 9 – pump cover; 10 – bypass valve; 11 – suction valve from the upper guide of the plunger bushing; 12 – plunger bushing; 13 – replaceable plugs; 14 – plunger; 15 – rotary sleeve for adjusting the supply advance; 16 – gear rack for adjusting the supply advance angle; 17 – toothed rack for adjusting the cyclic fuel supply; 18 – plunger reversal sleeve; 19 – sliding cu ffof the pusher; 20 – return springs; 21 – pushe r roller (adapted from [17, 22])

For preliminary review

Fuel is supplied to the injection pump through a pipeline through flange connections on the front wall of the housing from an electrically driven circulation pump. The fuel pressure in the low pressure circuit is maintained constant using a bypass valve located between the main fuel line and the fuel return line (Fig. 5.23).

Piston-type bypass valve with spring-loaded piston. The initial tension of the valve spring determines the pressure in the fuel supply line.

A bypass valve is installed on top of the pump cover, separating the pump discharge cavity from the drain line (Fig. 1.24). When the engine is running, the valve is in a closed state, delimiting the cavities. A piston pneumatic actuator is installed in the upper part of the valve housing, with the help of which the valve is held open through the rod. The valve is used to quickly (emergency) stop the engine, as well as when bleeding the system.

A suction valve is installed on the underside of the top cover (Fig. 1.25), the protruding housing of which serves as a guide for the plunger sleeve. During the suction stroke, the valve, under the influence of pressure in the pump cavity, overcomes the force of the retaining spring and opens, as a result of which the space above the plunger is filled with fuel.

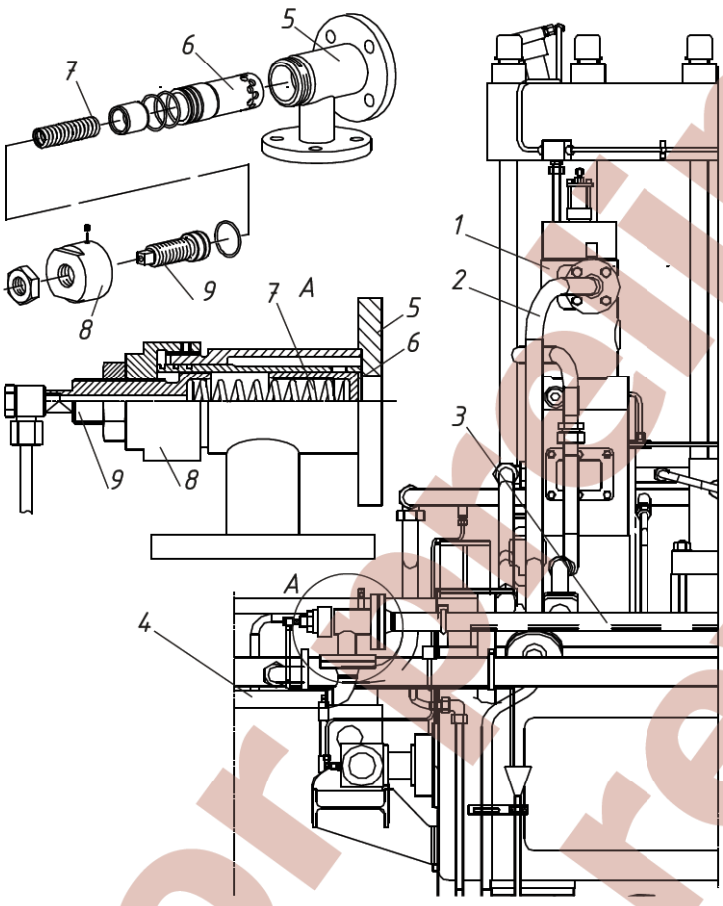


Figure 5.23 – Bypass pressure control valve: 1 – injection pump; 2 – fuel supply line to the injection pump; 3 – pressure line; 4 – drain line; 5 – valve housing; 6 – piston; 7 – spring; 8 – housing cover; 9 – pressure adjustment bolt (adapted from [17])

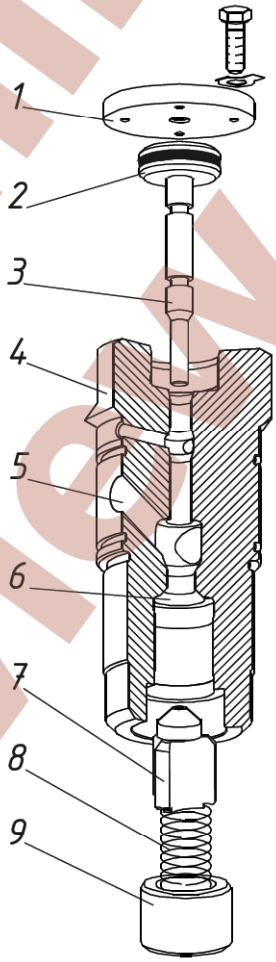


Figure 5.24 – Bypass valve of the diesel injection pump of the MC series from MAN: 1 – cover; 2 – pneumatic piston; 3 – rod; 4 – valve housing; 5 – bypass channel; 6 – valve seat; 7 – valve; 8 – spring; 9 – plug (adapted from [17])

The plunger and bushing are precisely adjusted to each other and form a precision pair that should not be disassembled during repair work. Symmetrically inclined cut-off edges with adjustment at the end of the supply are cut on the housing of the plunger. The symmetrical arrangement of the edges helps balance the lateral forces resulting from the pressure in the cavity under the edges. At the bottom of the plunger there is a guide collar of square cross-section, which slides into the slot

For preliminary review

of the rotary adjusting sleeve. At the lower end of the plunger there is a base with which it rests on the heel of the pusher and is fixed relative to it using an annular recess. A small gap between the base of the plunger and the pusher ensures free rotation of the plunger.

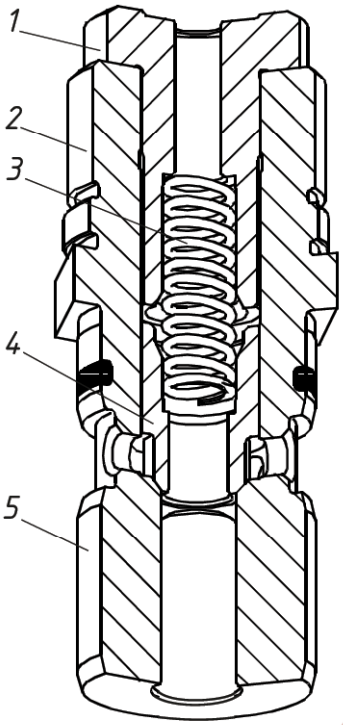


Figure 5.25 – Injection pump suction valve for a MAN MC series diesel engine: 1 – spring guide; 2 – valve housing; 3 – spring; 4 – valve; 5 – plunger bushing guide (adapted from [17])

The plunger sleeve is guided at the top and bottom of the pump housing. To seal between the bushing and the housing, there are three grooves on the outer surface of the bushing in which low-friction O-rings are installed. There is a drain hole in the lower sealing collar of the bushing, between the two O-rings. Rotation of the bushing is prevented by a guide screw installed on the front side of the injection pump housing.

To discharge fuel at the end of the injection stroke of the plunger, the bushing has two symmetrical shut-off channels made in the form of divergent pipes. This form makes it possible to reduce the resistance to liquid flow in the channel, reduce the rate of fuel outflow, thereby reducing the cavitation effect of the flow on the channel walls. Two plugs are installed opposite the bushing cut-off holes. Fuel jets at the end of the injection stroke under high pressure hit the plugs, causing their intense erosion. If the plugs are significantly worn, they can be replaced with new ones.

For axial movement of the bushing, a thread is cut at its lower end, which enters the internal thread of the rotary bushing for adjusting the supply advance (Fig. 5.22 *d*).

The rotary advance bushing has a toothed ring, with which the upper gear rack at the base of the fuel injection pump housing engages. The rack is connected to a pneumatic cylinder, the position of which is determined by the engine governor. Rotating the rotary sleeve causes the plunger sleeve to rise or fall, changing the position of the shut-off channels relative to the top edge of the plunger. Thus, moving the top rail leads to a change in the supply advance angle.

The cyclic supply is changed by turning the plunger using the lower gear sector and the rack.

To smooth out fuel pulsations that occur during the opening of the shut-off holes, a damper (compensator), which is a piston-type spring pressure accumulator, is attached to the flange on the rear wall of the pump housing (Fig. 5.26).

The pump is driven from the camshaft through a roller tappet installed in a separate housing. The pusher housing is secured to the camshaft housing with four studs, the threads on two of which have a length that allows for gradual weakening of the pusher spring during its dismantling.

Rotation of the cylindrical guide of the pusher is prevented by means of a locking pin installed in the housing bushing.

The pusher of each fuel pump includes a reversible link with angular movement. Each cylinder is reversed separately. The reversing mechanism is driven by compressed air.

For preliminary review

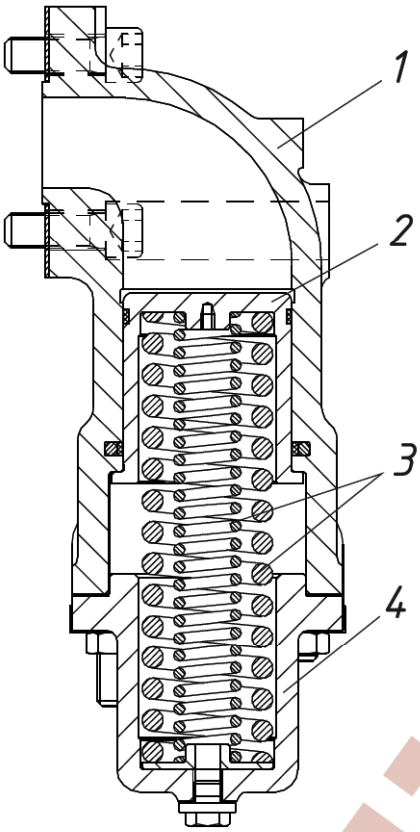


Figure 1.26 – Spring pressure pulsation damper: 1 – housing; 2 – piston; 3 – springs; 4 – cover (adapted from [17])

Reversing is carried out by moving the roller of the fuel injection pump drive mechanism of each cylinder. The link connecting the pusher and the roller has a reversing lever, at the upper end of which a pin is installed (Fig. 5.27), which moves in a guide attached to the control pneumatic cylinder.

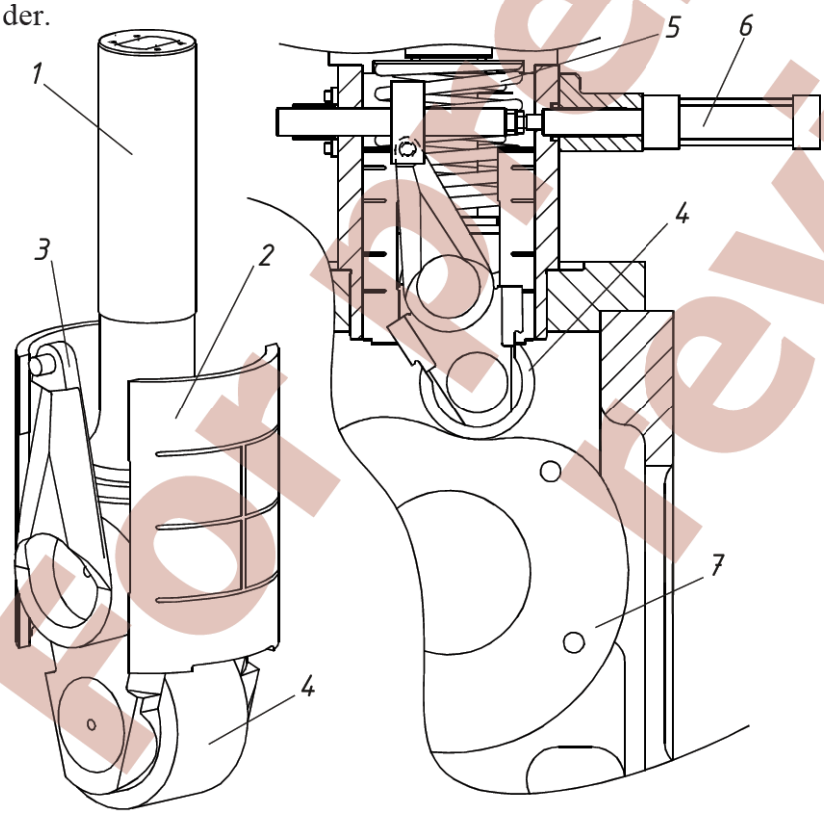


Figure 5.27 – Pusher with a reverse mechanism for a diesel engine of the MC series from MAN: 1 – pusher rod; 2 – guides; 3 – roller shift lever; 4 – pusher roller; 5 – return springs; 6 – pneumatic reverse cylinder; 7 – fuel injection pump drive cam (adapted from [17])

The reversing link is self-locking in both the «Forward» and «Backward» positions without the help of external forces. When the reversing link is moved from one position to another, the fuel supply angle changes relative to the TDC of the corresponding working cylinder, providing an optimal supply advance angle for both forward and reverse engine operation.

For preliminary review

A special cam is used to drive the roller tappet, which ensures optimal fuel supply regardless of the direction of engine rotation. The profile of the drive cam is shown in Fig. 5.28.

The downward movement of the pusher and the pressing of its roller against the cam is ensured by two spiral springs fixed between the pusher and the base of the pump.

The top of the annular groove of the pusher is located inside the pump base and is equipped with a cap. This cap, together with the sealing sleeve, which is hot pressed into the base of the pump, forms a labyrinth that prevents fuel from entering the camshaft lubrication system (Fig. 5.22 *b*).

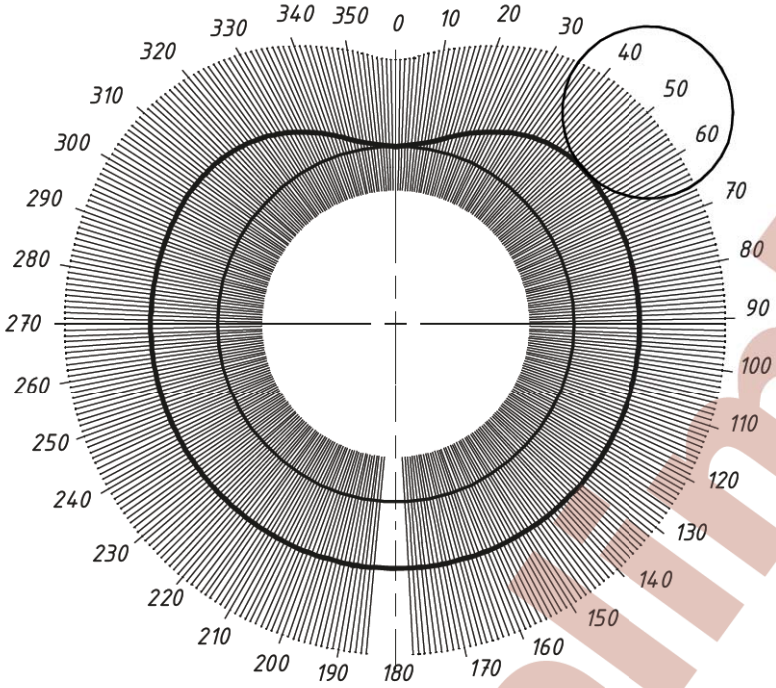


Figure 5.28 – Profile of the fuel cam of diesel engines of the MC series from MAN (adapted from [17])

Each tappet housing has an eccentric-type lifting and locking device that can be used to lift and lock the tappet over the fuel cam. The lifting device is installed on the side of the pusher housing (Fig. 5.29).

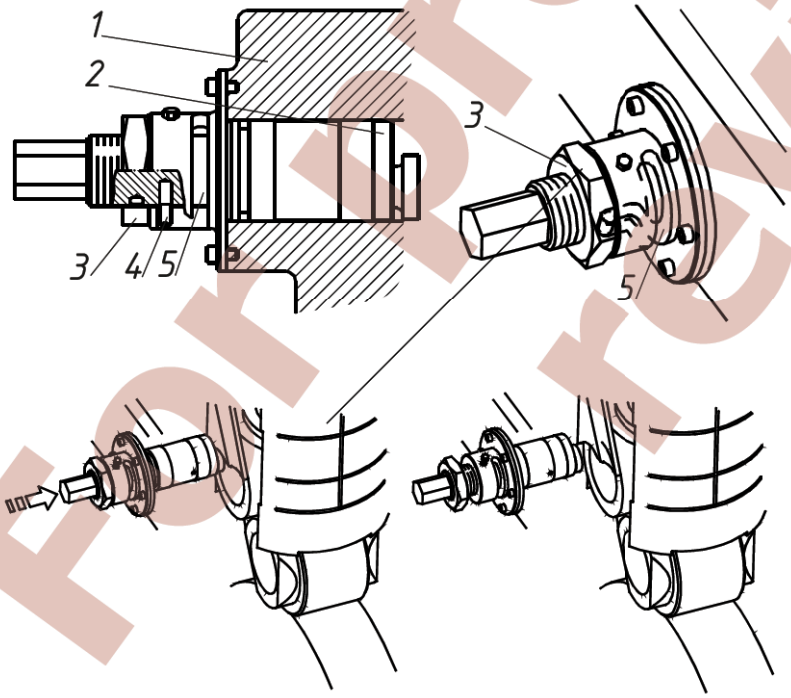


Figure 5.29 – Lifting and fixing device of the fuel injection pump pusher of a MAN MC series diesel engine of eccentric type: 1 – pusher housing; 2 – eccentric lift; 3 – lock nut; 4 – guide pin; 5 – guide groove; 6 – locking screw (adapted from [17, 26])

To raise the pusher, the lock nut on the eccentric shaft is released, and the eccentric protrusion is inserted under the reverse lever by axial movement. Next, by turning the eccentric, the pusher is raised and its position is fixed using a locking screw.

Otherwise, the problem of regulating the moments of the beginning and end of the supply is

For preliminary review

solved in the fuel injection pump of the 64 series engines from Wärtsilä. In these engines, single-piston pumps are replaced by double-piston pumps, in which one plunger controls the start of fuel supply, and the other controls the end. In this case, both plungers provide the required amount of the cyclic portion of fuel, since they have a parallel drive from the cam shaft. A general view of the pump is shown in Fig. 5.30, and the diagram of its operation is shown in Fig. 5.31.

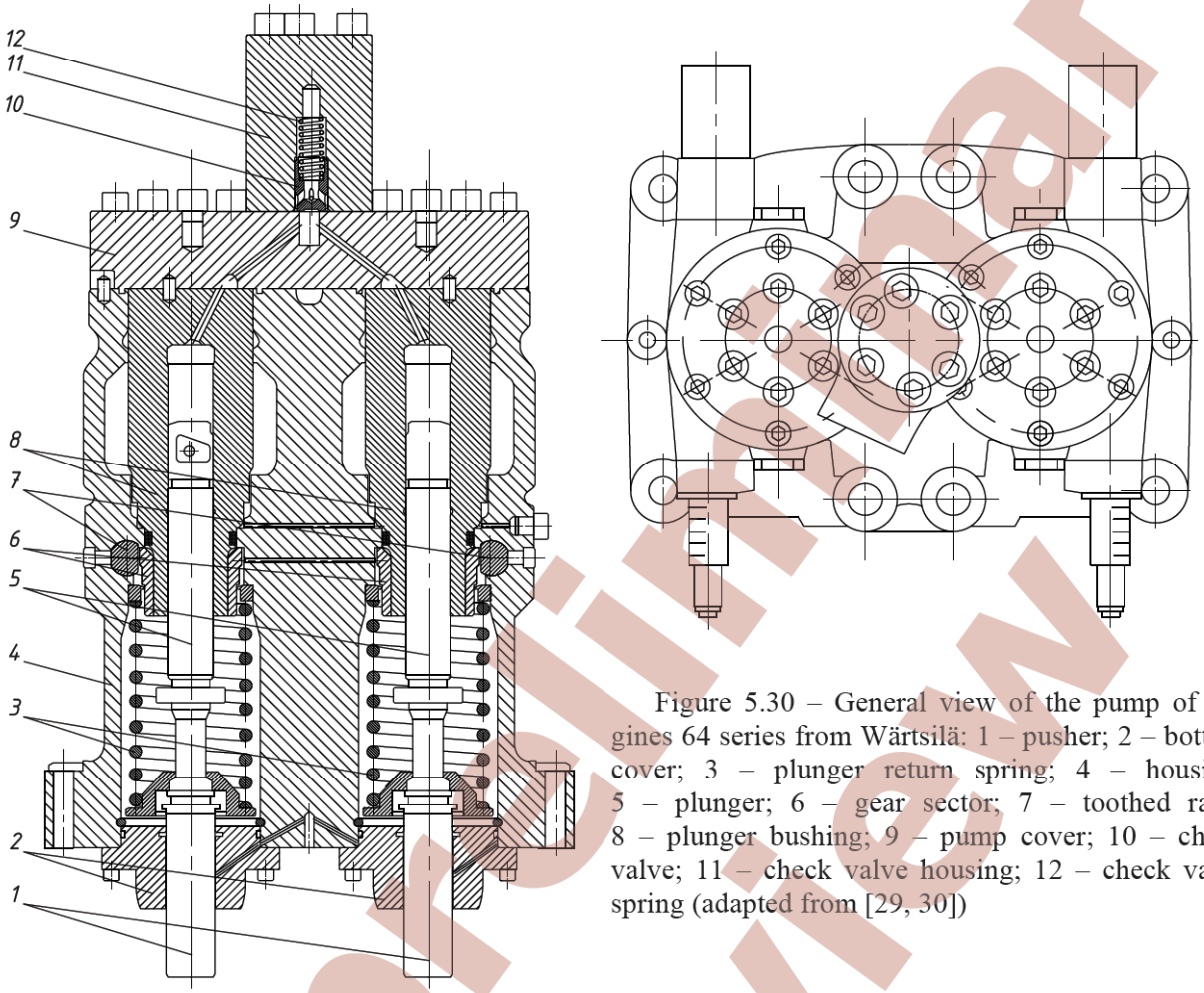


Figure 5.30 – General view of the pump of engines 64 series from Wärtsilä: 1 – pusher; 2 – bottom cover; 3 – plunger return spring; 4 – housing; 5 – plunger; 6 – gear sector; 7 – toothed rack; 8 – plunger bushing; 9 – pump cover; 10 – check valve; 11 – check valve housing; 12 – check valve spring (adapted from [29, 30])

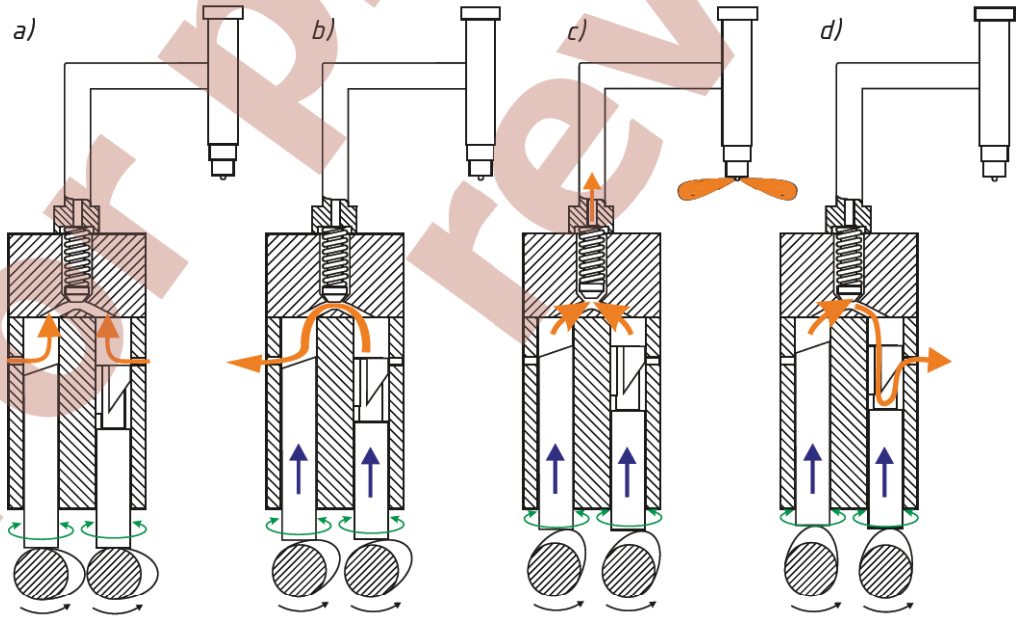


Figure 5.31 – Double-plunger injection pump of the L64 engine from Wärtsilä: *a* – filling the pump; *b* – the plunger that controls the supply closes the shut-off port, excess fuel leaves through the filling port; *c* – both ports are closed, fuel supply to the injector begins; *d* – the cut-off port is aligned with the lower edge of the cut-off edge, the supply stops (adapted from [30])

For preliminary review

Independent control of the plunger rotation mechanisms allows you to set the supply advance angle regardless of its value, optimize the supply law depending on the engine operating mode by redistributing the values of the active strokes of the plungers.

The use of two plungers instead of one makes it possible to reduce their diameter, reduce mechanical loads in the drive mechanism, increase the rigidity of the plunger and bushing, which, in turn, reduces the gaps in the plunger pairs and, consequently, reduces leakage through them.

5.4.5 Pump-injectors with spool control

Spool-controlled pump-injectors are not fundamentally different from fuel pumps in terms of the method of regulating the cyclic supply and the start point of fuel injection. The only difference is that the pump section, together with the mechanism for turning the plunger, is mounted in the same housing with the fuel injector. As a result, the design is more compact, which allows the internal volume to be minimized and its harmful effect on the fuel supply process reduced. Figure 5.32 shows the design of a high-speed diesel pump-injector with end-of-supply control from General Motors.

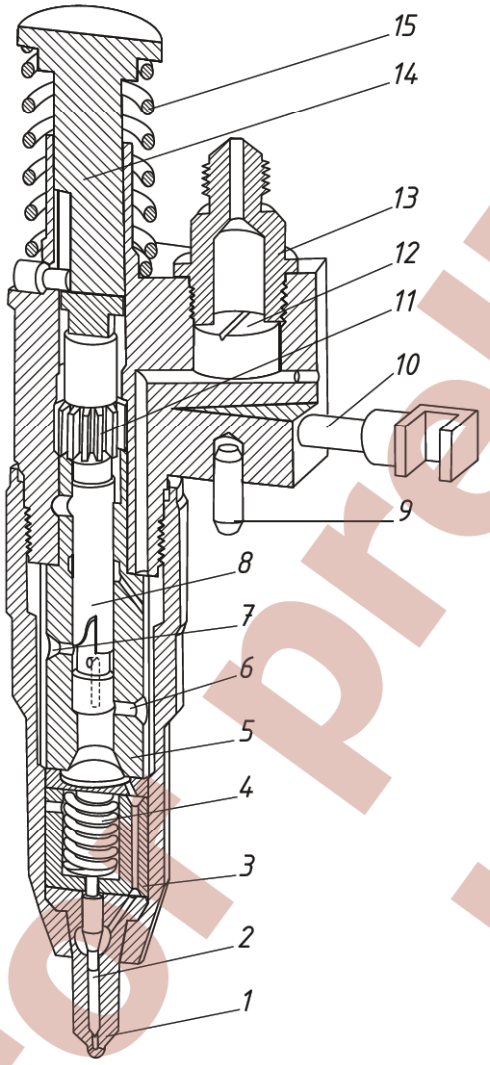


Figure 5.32 – General Motors high-speed diesel pump-injector: 1 – fuel injector housing; 2 – needle valve of the nozzle; 3 – spacer; 4 – needle valve spring; 5 – plunger bushing; 6 – filling hole; 7 – drain hole; 8 – plunger; 9 – guide pin; 10 – toothed rack; 11 – gear sector; 12 – fuel filter; 13 – connecting fitting; 14 – pusher; 15 – pusher return spring (adapted from [31])

The plunger is rotated using a toothed rack and sector. A significant disadvantage of this arrangement is the presence of an individual rack drive for each pump-injector, which somewhat complicates the synchronization of their joint operation.

With a relatively small internal volume of space between the plunger and the fuel injector, the influence of pressure waves is not enough to re-open the needle valve of the fuel injector and cause after-injection, so most injectors do not have pressure valves. On larger engines, an increase in the dimensions of the pump-injectors has led to an increase in the internal volume, so a delivery valve is installed between the cavities of the plunger and the fuel injector.

For preliminary review

5.5 Fuel pumps and valve-controlled pump-injectors

With valve regulation, the magnitude of the active stroke of the plunger, as well as the moments of the beginning and end of fuel supply, depend on the position of the valves that disconnect or communicate the space above the plunger with the low-pressure cavity. In marine diesel engines, valves are usually mechanically driven, but some designs of modern pumps and pump-injectors use an electromagnetic valve drive.

5.5.1 Fuel pumps with mechanically driven valves

The most general diagram of a fuel pump with valve-controlled supply and mechanical valve drive is shown in Fig. 5.33.

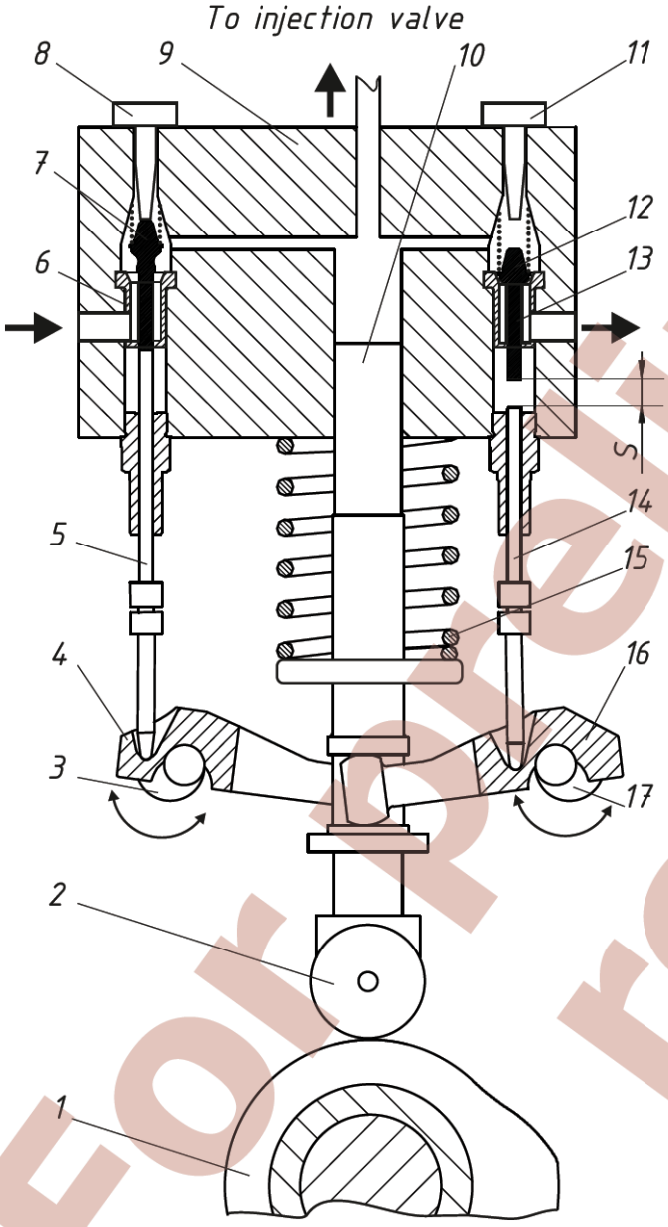


Figure 5.33 – Injection pump diagram with valve regulation and mechanical valve drive: 1 – cam shaft; 2 – pusher roller; 3, 17 – eccentric; 4 – suction valve drive lever; 5, 14 – push rod; 6 – suction valve; 7, 12 – valve spring; 8, 11 – valve stroke limiter; 9 – housing; 10 – plunger; 13 – drain valve; 15 – plunger return spring; 16 – drain valve drive lever (adapted from [9])

Depending on the design, pumps of this type can be equipped with one or two valves. In pumps equipped with one valve, regulation can be carried out at the end or beginning (Fig. 5.34 *a*, *b*), and in pumps with two valves – both at the beginning and at the end of the flow (Fig. 5.34 *c*). In pumps and pump-injectors with solenoid valve control, both the start and end of the flow can be changed in any combination.

Let us consider in more detail the operation of the pump shown in Fig. 5.33. The above-plunger space of the pump communicates with the low-pressure cavity through the filling cavity, and with

For preliminary review

the drain cavity through the drain valve. The geometric start of injection is determined by the moment of closing the suction valve, and the geometric end of injection depends on the moment of opening of the drain valve. The opening moments of the valves are determined by the size of the gaps *S* in the drive mechanism, which are regulated by turning the eccentric shafts (Fig. 5.33).

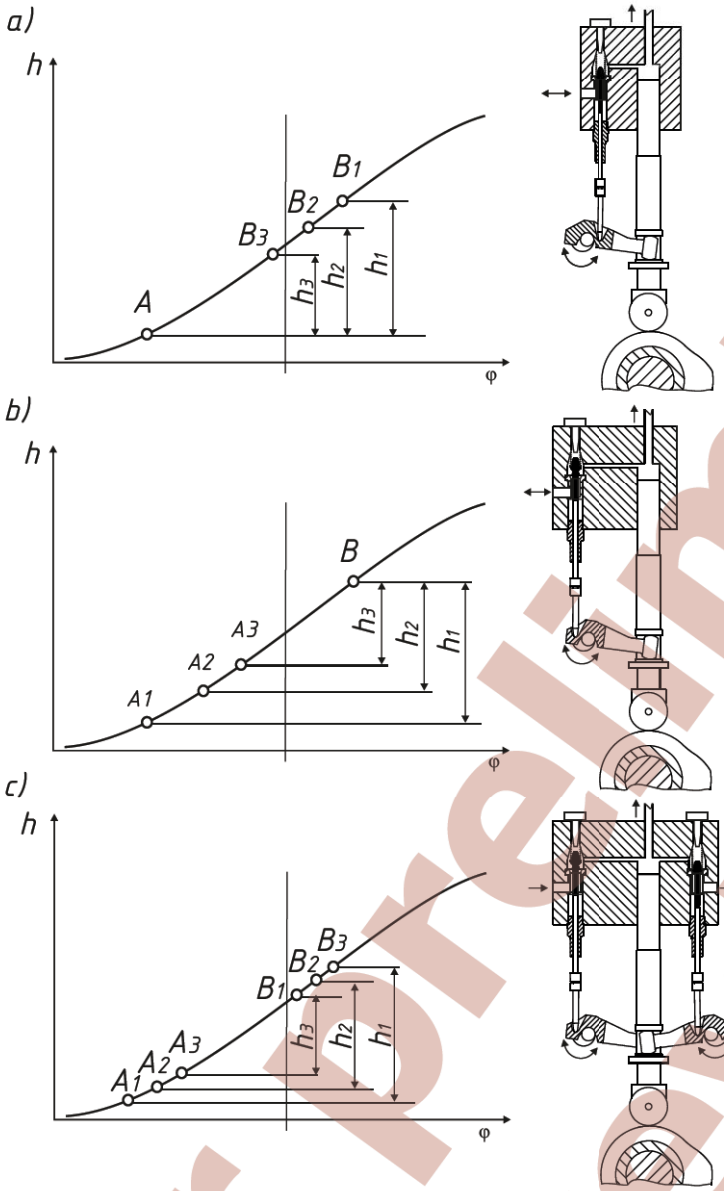


Figure 5.34 – Valve control of fuel supply: *a* – at the end of supply; *b* – at the beginning of the supply; *c* – at the beginning and end of the supply (adapted from [9])

A general view of the pump block, which includes the injection pump, is shown in Fig. 4.31, 5.35, and a cross section of the pump block along the injection pump axis is shown in Fig. 5.36 *a*. The same figure shows the individual elements of the pump design.

The valves are driven through a system of levers and rods from a recess on the bottom of the plunger. The suction valve is actuated through a double-arm lever, and the drain valve is operated through a single-arm lever. The eccentric shafts of the levers are rotated by the engine speed controller, which controls the advance angle of the start of supply, the angle of the end of injection and, consequently, the duration of fuel supply to the cylinder, on which the size of the cyclic portion of fuel directly depends.

The advantages of injection pump with valve regulation include:

- the absence of adjusting edges on the surface of the plunger, which weaken it and create lateral forces that accelerate wear;

- in pumps of this type, the principle of adjusting the moments of the beginning and end of flow is easier to implement than in pumps with spool control. At the same time, they contain fewer precision pairs.

For preliminary review

Figure 5.35 – Block of fuel supply pumps and actuators for the hydraulic drive of the exhaust valves of the RTA diesel engine from Wärtsilä (adapted from [25])

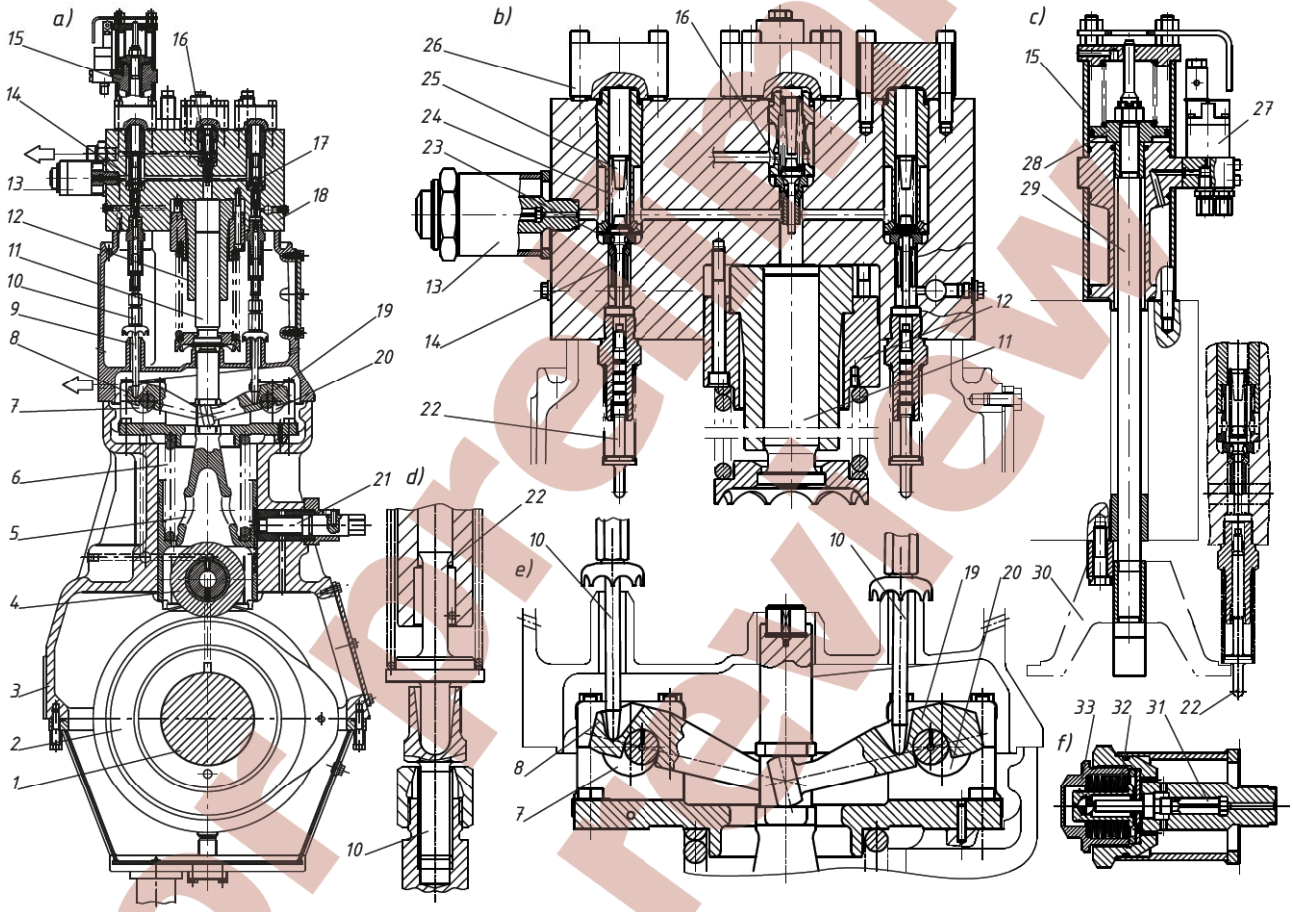


Figure 5.36 – High pressure section with valves control of cyclic supply and its elements (diesel RTA58T from Wärtsilä): *a* – general view of the injection pump; *b* – pump cover with valves; *c* – engine emergency stop mechanism; *d* – valve drive pusher rod; *e* – valve drive mechanism; *f* – safety valve. 1 – pump block shaft; 2 – plunger drive cam; 3 – pump housing; 4 – pusher roller; 5 – pusher; 6 – return spring; 7 – eccentric shaft of the suction valve drive levers; 8 – suction valve drive lever; 9 – push rod; 10 – mechanism for adjusting the gaps in the valve drive; 11 – plunger; 12 – plunger bushing; 13 – safety valve; 14 – suction valve; 15 – emergency fuel shut-off mechanism; 16 – discharge valve; 17 – drain valve; 18 – pump cover; 19 – drain valve drive lever; 20 – eccentric shaft of the drain valve drive levers; 21 – pusher fixation device; 22 – valve pusher; 23 – valve seat; 24 – valve return spring; 25 – valve travel limiter; 26 – valve well cover; 27 – control mechanism for the emergency stop valve; 28 – pneumatic piston of the emergency stop mechanism; 29 – rod of the emergency stop mechanism; 30 – traverse for lifting suction valves; 31 – safety valve needle; 32 – disc spring of the safety valve; 33 – safety valve adjusting cap (adapted from [16])

For preliminary review

Disadvantages include large dimensions, a more complex mechanical design, and low reliability of valve pairs. Especially in difficult conditions are drain valves that operate at high pressures and initial fluid flow rates. This leads to cavitation processes occurring at the moment the valve opens, which contribute to its rapid destruction. Partially, the problem of reliability of valve pairs is solved by installing them in the form of separate units, which can be easily replaced with new ones.

Currently, valve-type fuel injection pumps with mechanical valve drives are used only in diesel engines of the RTA series, which was developed by Sulzer and inherited by Wärtsilä.

To stop the engine, an emergency stop mechanism with a pneumatic drive is installed on the pump cover. The air to power the mechanism comes from the engine control system. When the «Stop» command is received, the pneumatic piston moves up and lifts the crossbar through the rod, which, in turn, lifts the suction valves and locks them in the open position (Fig. 5.36 *c*). The supra-plunger cavity is connected to the low-pressure cavity, and fuel injection stops.

To protect the pump from excessive pressures, a safety valve is installed in the cavity above the plunger (Fig. 5.36 *f*).

The fuel pump is driven by a cam with a symmetrical profile (Fig. 5.37).

This makes it possible to ensure the same flow law when the engine is running in forward and reverse gear.

The cams are located on couplings sliding relative to the drive shaft, which are attached to the stator of the reverse mechanism servo drive located on the drive shaft. A rotary-type hydraulic mechanism is used as a servo drive (Fig. 5.38), the rotor of which is formed by two gates mounted on the pump shaft.

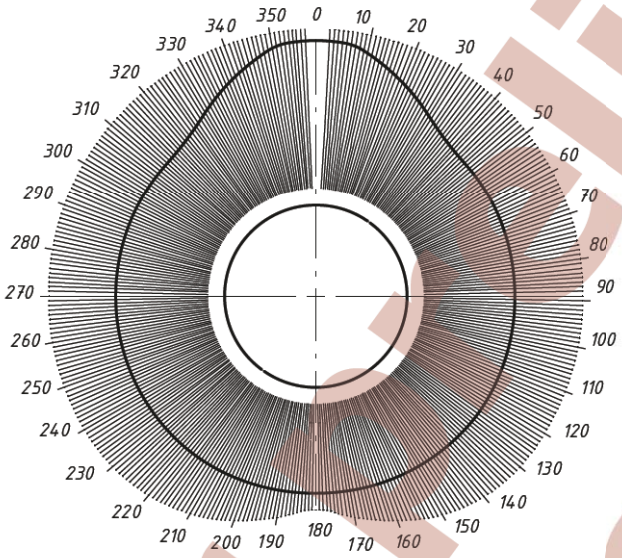


Figure 5.37 – Profile of the fuel cam of the injection pump of RTA series engines from Wärtsilä (adapted from [16])

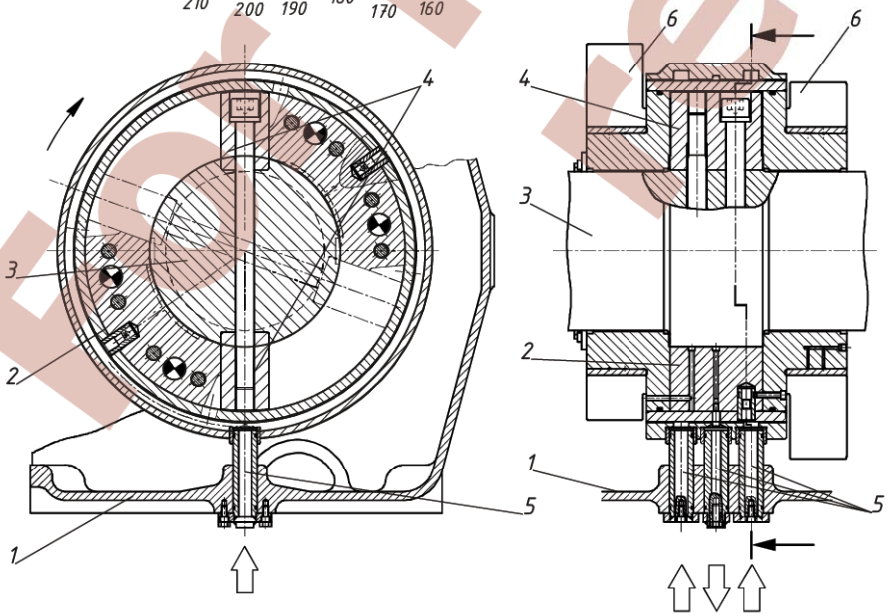


Figure 5.38 – Servo drive of the reverse mechanism of diesel engines of the RTA series from Wärtsilä: 1 – pump housing; 2 – stator; 3 – drive shaft; 4 – gate; 5 – control oil supply tubes; 6 – injection pump drive cams (adapted from [25])

Preliminary review

The drive of the eccentric shafts that control the start and end of the supply is carried out from the speed controller through a special drive mechanism, which implements the supply start angle control algorithm (VIT). The diagram and general view of the drive mechanism are shown in Fig. 5.39. Through a system of rods and levers, the regulator affects not only the amount of cyclic supply, but also the advance angle of fuel injection. The latter changes when the fuel supply changes and is achieved by rotating the eccentric that controls the suction valve drive lever.

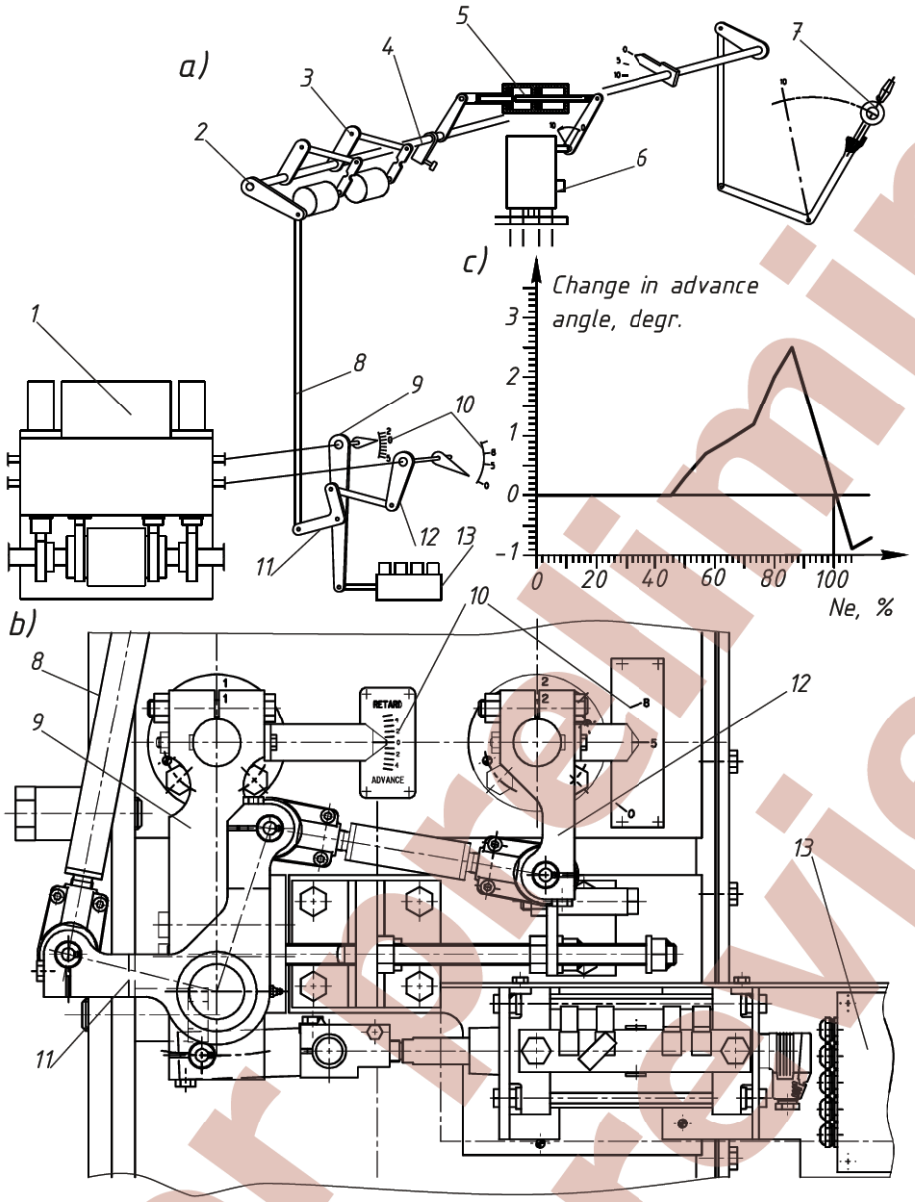


Figure 5.39 – Diagram (*a*) and general view (*b*) of the drive mechanism of the eccentric fuel supply control shafts of the injection pump of RTA series engines: *c* – graph of the change in the advance angle depending on the engine load. 1 – injection pump; 2 – drive shaft; 3 – load sensor drive levers; 4 – supply limiter; 5 – amplifier; 6 – speed regulator; 7 – remote control handle; 8 – traction; 9 – drive lever for the suction valve control shaft; 10 – scale for controlling the angle of the beginning and end of the supply; 11 – intermediate lever; 12 – drive lever for the drain valve control shaft; 13 – pneumatic cylinder (adapted from [32])

Mechanical transmission from the regulator to the eccentric shafts is carried out through an amplifier in such a way that in the range of engine loads from 75 to 100%, the supply advance angle increases (Fig. 5.40). This allows you to maintain the maximum cycle pressure at a given constant level, which has a positive effect on the fuel efficiency of the engine.

When the load decreases below 75% of the nominal value, maintaining the specified maximum cycle pressure becomes impractical, since in this case the rigidity of the operating process, characterized by the rate of pressure increase, increases. An increase in rigidity leads to an increase in shock loads on the crankshaft parts. To reduce the rigidity of the operating process in partial load modes, a reduction in the injection advance angle is provided (Fig. 1.40).

In addition, a decrease in the injection advance angle leads to the fact that fuel supply begins at a higher speed section of the plunger movement. As a result, the quality of atomization improves and the completeness of fuel combustion increases.

For preliminary review

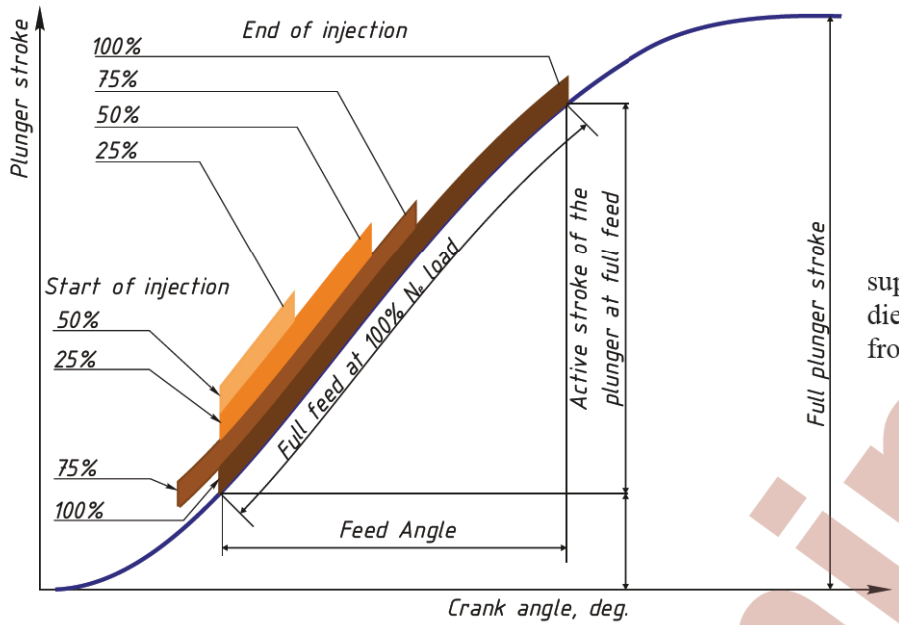


Figure 5.40 – Phases of fuel supply to the injection pump of diesel engines of the RTA series from Wärtsilä (adapted from [33])

5.5.2 Fuel pumps and pump-injectors with solenoid valve actuators

This type of fuel pumps and pump-injectors combines the advantages of valve-controlled pumps and allows the fuel supply process to be controlled within a wide range using controllers based on microprocessor technology. Optimization of the fuel supply law is carried out by a program embedded in the electronic control unit, which receives and processes signals from sensors of rotation speed, load, boost pressure, fuel temperature, etc. Based on the processing of the received signals, the program determines the optimal values for the start and end of fuel supply for a given mode and applies voltage to the control valve drive solenoids. When using pumps with solenoid controlled valves, fuel injection can be carried out through a traditional injector design, which greatly simplifies the implementation of such systems on diesel engines. The only complex and non-traditional element of such injection pumps is the control valve, which is subject to a number of specific requirements. U in high-speed diesel engines, the valve must operate in both directions in a time of no more than 0.1...0.2 ms, and this is possible with electromagnet forces of at least 250 N, and then only with small masses of moving parts.

Figure 5.41 shows a pump-injector with a solenoid actuated control valve, type EI800, which is equipped with Caterpillar 3500 and 3600 series engines.

The operating diagram of the EI800 type pump-injector is shown in Fig. 5.42.

When the plunger moves upward, the space under the plunger is filled. Fuel from the low pressure line enters through the open control valve through the bypass channel into the cavity under the plunger (Fig. 5.42 *a*).

As the plunger moves downward (discharge stroke), the pressure underneath it will not increase as long as the control valve is in the open position.

When voltage is applied from the control unit to the drive solenoid, the valve rises and sits on the seat, as a result of which the cavity under the plunger will be disconnected from the low pressure line. The pressure in the cavity under the plunger begins to increase, and when it reaches 35 MPa, the fuel injector needle, overcoming the force of the spring, rises and fuel injection into the combustion chamber begins (Fig. 5.42 *b*).

When the power to the solenoid is turned off under the action of a spring, the valve opens and the fuel supply to the cylinder stops (Fig. 5.42 *c*), then the plunger idles, and then the whole cycle repeats again.

This system allows you to supply a cyclic portion of fuel both in one injection and to split the cyclic portion into several successive injections.

Cummins used its well-developed scheme using open-type pump-injectors, adapting their design to the electronic engine control system. This fuel system is widely used on high-speed engines, which are used on merchant ships mainly to drive diesel generators.

For preliminary review

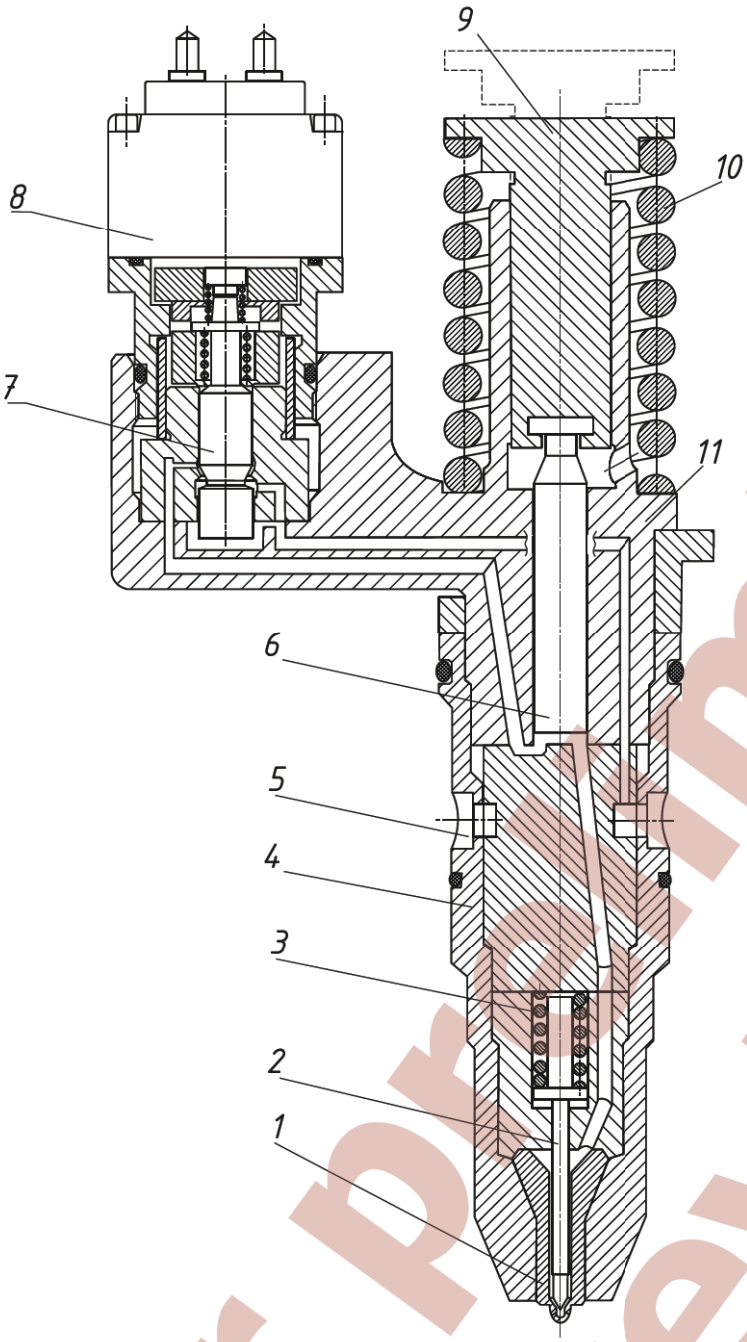


Figure 5.41 – Pump-injector with an electromagnetic drive of a control valve type EI800: 1 – fuel injector housing; 2 – needle valve; 3 – needle valve spring; 4 – housing; 5 – low pressure line; 6 – plunger; 7 – valve; 8 – solenoid; 9 – pusher; 10 – pusher spring; 11 – bushing (adapted from [9, 34])

The system consists of pump-injectors and a fuel supply control unit.

The control unit includes booster pumps, solenoid valves for fuel dosing and start-of-supply control, control sensors, and an electronic engine control unit. All of the listed elements are mounted in the form of one module, which is directly mounted on the engine (Fig. 5.43). The electronic control unit is cooled by a flow of fuel, part of which (≈ 35%) after the booster pump enters a tubular coil mounted directly under the housing of the microprocessor unit.

A distinctive feature of this system is that to supply fuel to the combustion chamber, an open pump-injector with a mechanical drive and hydraulic control of the start and duration of injection is used.

This type of pump-injectors is distinguished by the fact that they do not have a needle valve that separates the internal space of the nozzle and the combustion chamber throughout the entire cycle, with the exception of the injection process. The fuel enters directly under the plunger, which at the end is made in the form of a shut-off cone, which also performs the function of a displacer, which, with its volume, ensures complete displacement of fuel from the internal cavity of the nozzle.

For preliminary review

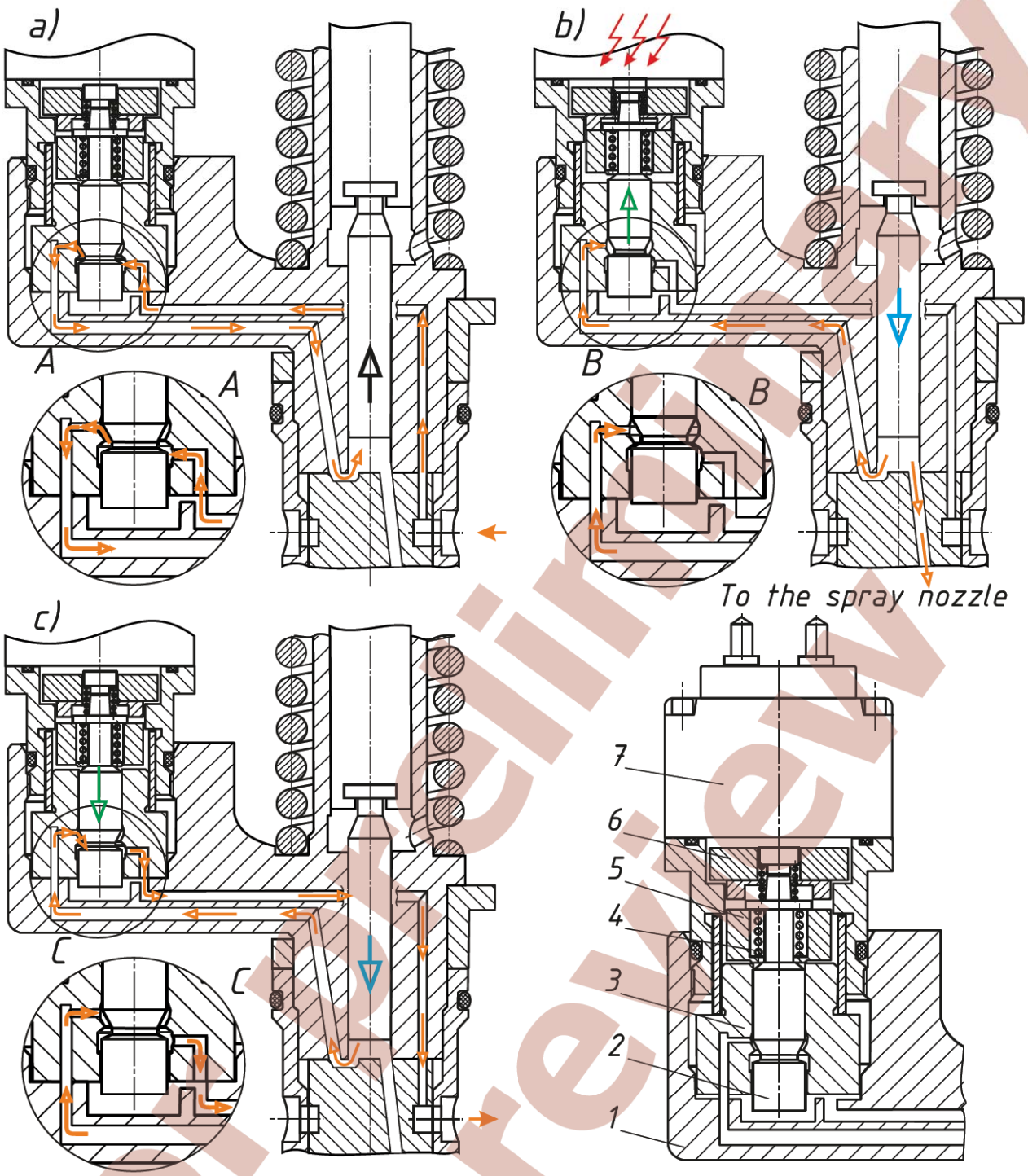


Figure 5.42 – Stages of the injection process of an electrically controlled pump-injector with supply valve control and the main elements of the electromagnetic control valve: *a* – filling the sub-plunger space; *b* – discharge stroke of the plunger; *c* – idling stroke of the plunger; *d* – supply control solenoid valve. 1 – solenoid valve housing; 2 – valve; 3 – valve seat; 4 – return spring; 5 – armature spring; 6 – anchor; 7 – solenoid (adapted from [9])

When filling the sub-plunger space with a new portion of fuel, its flow through the fuel injector holes is prevented by excess pressure in the combustion chamber, which during the compression stroke exceeds the pressure in the fuel injector cavity.

The general view of the pump-injector is shown in Fig. 5.44.

The pump-injector housing houses three plungers: upper, intermediate and lower, injection, which, in fact, ensures fuel injection into the combustion chamber.

For preliminary review

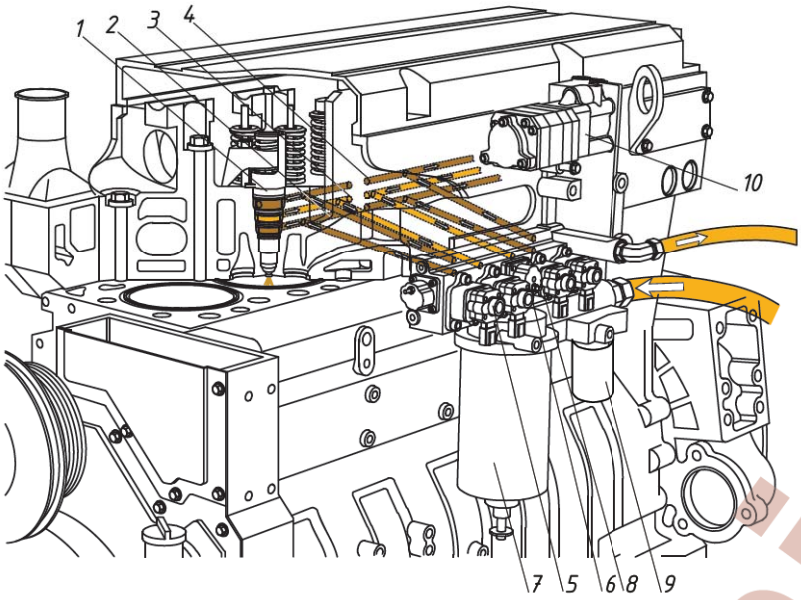


Figure 5.43 – Location of fuel system elements on the Cummins CM870 engine: 1 – pump-injector; 2 – fuel supply channel for injection; 3 – channel for draining fuel; 4 – channel for supplying fuel to the system for regulating the start of supply; 5 – valve for dosing the cyclic portion of fuel; 6 – injection start control valve; 7 – fuel filter; 8 – supply shut-off valve; 9 – fuel priming pump; 10 – fuel pump (adapted from [9, 35])

The lower plunger, together with the spray housing, forms the discharge cavity. A control cavity is formed between the upper and intermediate plungers. By changing the volume of fuel supplied to this cavity, you can change the start time and duration of injection.

Between the intermediate and injection plungers there is a drain cavity in which leaks are collected, as well as fuel used to control the start of delivery after the injection of the main portion has ended.

All cavities of the pump-injector communicate through a system of valves and grooves on the housing with the corresponding channels for supplying and discharging fuel, which are made in the cylinder head. Each group of channels serves three cylinders.

A measured portion of fuel is supplied from the control unit to the fuel supply channel for injection. Since at this moment only one pump-injector out of three is in the open state, this portion enters exactly this pump-injector.

The same thing happens with a measured portion of fuel coming from the control channel for the start of fuel supply.

Dosing of the cyclic and control portions of fuel is carried out using solenoid valves controlled by an electronic unit.

The general diagram of the fuel supply control system is shown in Fig. 5.45 *a*.

The electronic unit processes the information received from the sensors and generates a valve control signal. Each pair of valves simultaneously serves up to three pump-injectors. Structurally, both valves are absolutely identical and interchangeable (Fig. 5.45 *b*). In the absence of a control signal, the valves are in a closed state, in which the shut-off cone is pressed against the valve seat under the action of a spring. When a signal is applied, the solenoid core of the solenoid valve actuator is magnetized, the armature is attracted to the core, opening the valve. Fuel supplied to the valve under a constant pressure of 2.62 MPa, passing through it, enters the corresponding channel. The amount of fuel entering the channel is determined by the valve opening time.

When the pump-injector drive pusher is released, all plungers move upward under the action of return springs (Fig. 5.46 *a*).

With further movement, the pusher of the injection plunger rests against the limiter on the housing and the movement of the injection plunger stops.

The lower edge of the injection plunger opens the filling hole on the fuel injector housing, and fuel from the supply line, through the suction valve, fills the injection cavity of the fuel injector (Fig. 5.46 *b*). The degree of filling of the discharge cavity and, consequently, the cyclic flow are determined by the opening time of the metering control valve. After placing the discharge plunger on the stop, the intermediate and upper plungers continue to move upward until the cavity formed by the annular groove on the housing of the intermediate plunger is aligned with the filling hole of the supply start control line.

For preliminary review

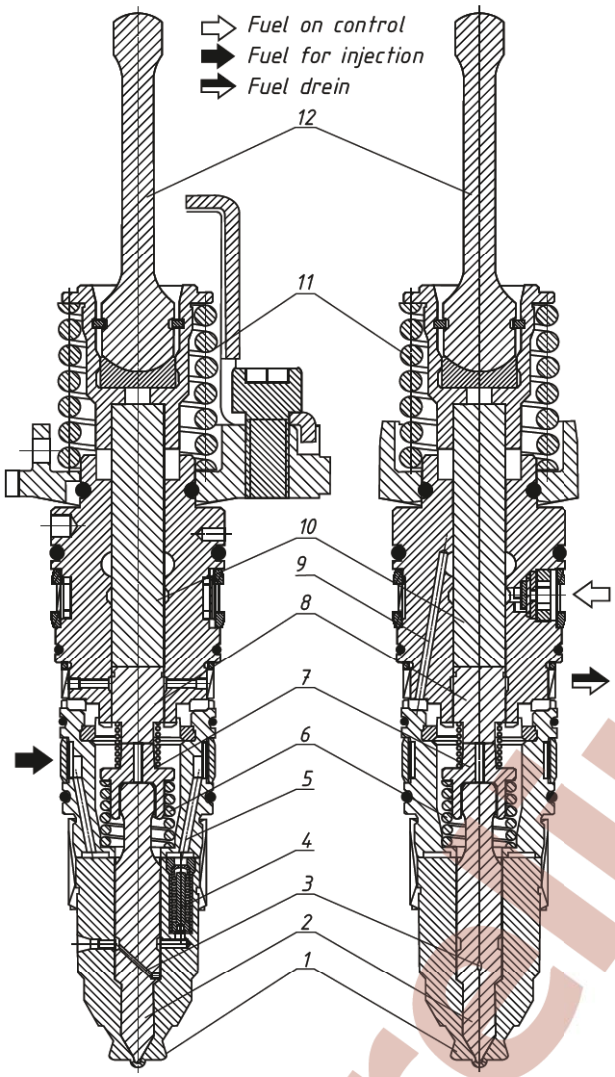


Figure 5.44 – Pump-injector of Cummins engines with an electronic supply control system type ISX: 1 – spray housing; 2 – injection plunger; 3 – drainage channel; 4 – suction valve; 5 – filling channel; 6 – intermediate spring; 7 – pusher of the injection plunger; 8 – intermediate plunger; 9 – leakage channel; 10 – upper plunger; 11 – return spring; 12 – pusher rod (adapted from [35])

The operating procedure of the pump-injector of the ISX system is shown in Fig. 5.46.

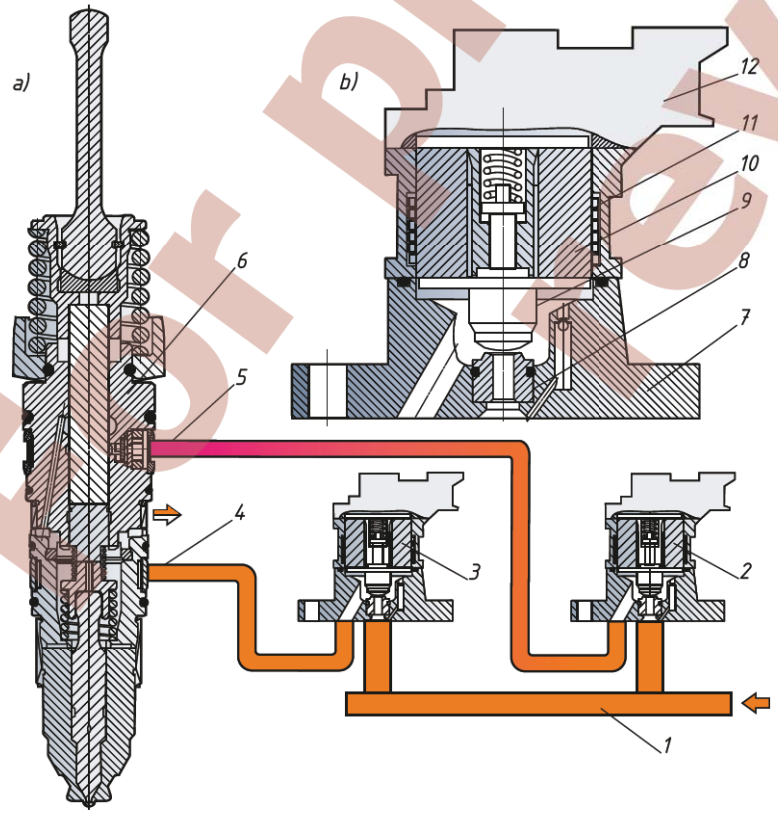


Figure 5.45 – General diagram of the fuel supply control in the Cummins ISX injection system (*a*) and control valve diagram (*b*): 1 – fuel supply line from the fuel pump; 2 – injection control valve; 3 – cyclic supply dosing valve; 4 – fuel supply channel for injection; 5 – channel for supplying fuel to the system for regulating the start of supply; 6 – pump-injector; 7 – valve housing; 8 – valve seat; 9 – shut-off cone; 10 – magnetic core; 11 – solenoid coil; 12 – electrical connector (adapted from [9, 35])

For Preliminary Review

Opening the injection start time control valve leads to the fact that this cavity begins to fill with fuel, which in this case is used as a control hydraulic fluid (Fig. 5.46 *c*).

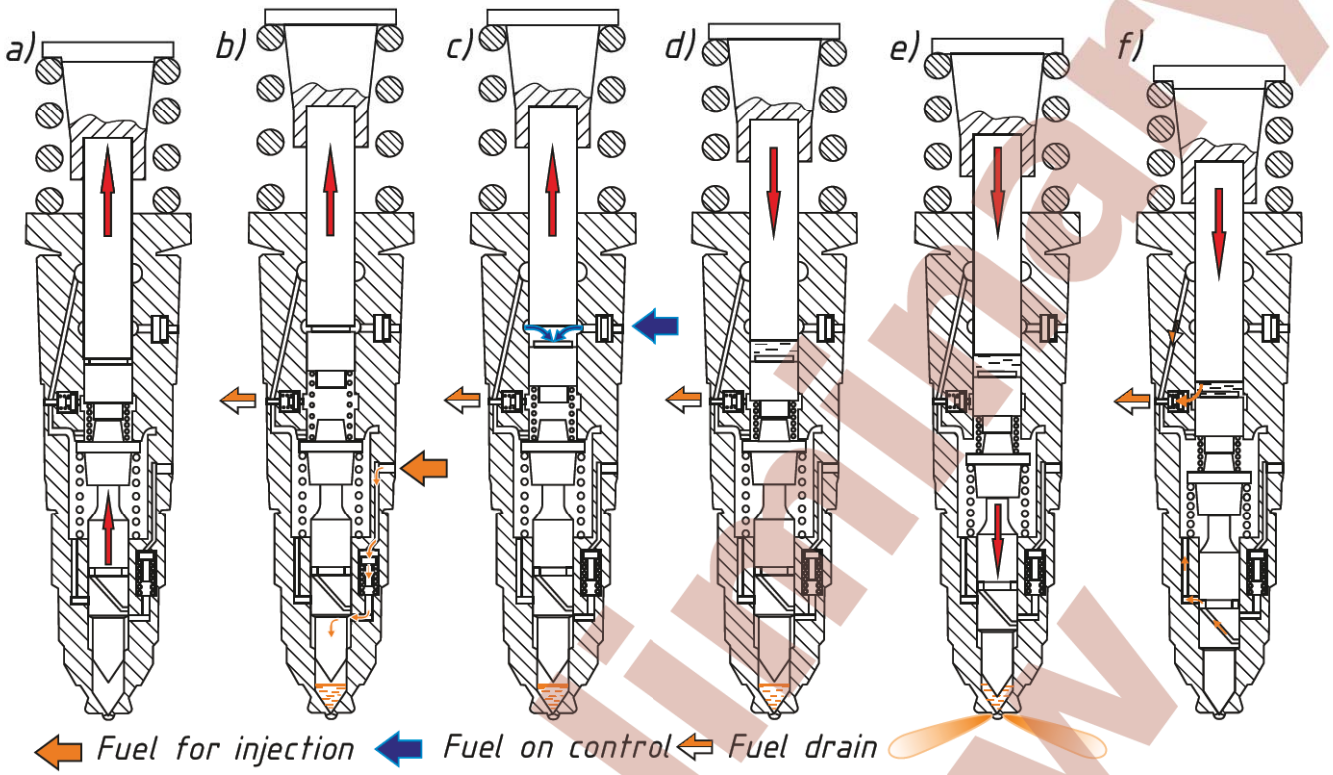


Figure 5.46 – Operating order of the ISX series pump-injector from Cummins: *a* – lifting the plungers to their original position; *b* – lifting the upper and intermediate plunger, filling the discharge cavity; *c* – filling the cavity for controlling the start time and duration of injection; *d* – downward movement of the upper and intermediate plungers; *e* – fuel injection; *f* – draining fuel from the discharge cavity and control cavity [9, 35]

Under the influence of fuel pressure, the intermediate plunger moves down by an amount proportional to the volume of incoming fuel. The volume, in turn, depends on the opening time of the solenoid valve for controlling the start of injection.

To prevent displacement of part of the fuel during the return stroke of the upper plunger, a plate-type check valve is installed between the supply channel and the control cavity.

When the upper plunger moves downwards, the closing of the plate valve, and then the closing of the filling hole by the edge of the plunger, leads to the formation of a closed volume filled with fuel between the upper and intermediate plungers. Through this volume, the force is transmitted from the upper plunger to the intermediate one. The size of this volume determines the distance and, therefore, the time required for the intermediate plunger to rest against the pusher of the discharge plunger (Fig. 5.46 *d*).

Thus, the volume of fuel supplied to the control cavity determines the moment the injection begins, and the average speed in the injection section determines its duration.

When the intermediate plunger rests against the discharge pusher, all three plungers begin their downward movement. The suction valve of the discharge cavity cuts off the reverse movement of fuel or its vapors into the filling line, and when the lower edge of the plunger closes the filling hole, the discharge cavity is completely disconnected from the filling line. Fuel injection begins into the engine combustion chamber (Fig. 5.46 *e*). The absence of intermediate volumes between the injection cavity and the fuel injector cavity makes it possible to obtain very high injection pressures from 190 to 240 MPa.

After the discharge plunger displacer reaches the bottom position, the annular groove on the intermediate plunger will align with the drain hole in the bushing. Fuel is forced from the control cav-

For preliminary review

ity through the drain valve into the drain line (Fig. 5.46 *f*). The remaining fuel from the injection cavity is also discharged into the drain line through the drainage channel in the injection plunger.

Then the whole cycle repeats.

The advantages of the considered fuel system scheme include:

- sufficiently high injection pressure, which promotes high-quality atomization and combustion of fuel;
- absence of solenoid valves on the injectors, which are in most cases a problematic design element;
- electromagnetic valves control the process of filling the discharge cavity, and not the injection itself, so there are no significant requirements for their speed (filling occurs over a fairly long time);
- one pair of solenoid valves controls three injectors at once, and the valves themselves are located in an easily accessible place and, if necessary, can be simply replaced with new ones without disassembling the engine.

The disadvantages of this scheme are:

- the complexity of the design of the pump-injectors themselves, primarily due to the presence of a large number of precision parts;
- the need for precise adjustment of all elements to ensure effective interaction between the mechanical, hydraulic, electrical and electronic components of the system.

5.6 Fuel systems with hydraulically driven fuel pumps

5.6.1 Fuel pumps for low-speed diesel engines with hydraulic drive

When developing new concepts in the field of organizing the operating processe of modern marine diesel engines, a number of manufacturers have taken the path of replacing standard injection pumps with fuel pumps with a hydraulic plunger drive and electronic fuel supply control. In the field of low-speed engines, such systems have been developed by MAN and Mitsubishi. Caterpillar is increasingly introducing electrically controlled hydraulically driven pump-injectors in its high-speed diesel engines.

The first LSE with a hydraulic fuel supply system was released by Mitsubishi in 2002 and received the designation UEC Eco-Engine.

In parallel with Mitsubishi, MAN, based on MC series engines, developed its own version of the fuel injection system with an electrically controlled hydraulic drive. The company has been producing engines equipped with this system since 2003 under the symbol ME. In addition to the fuel supply in engines of this series, the electronic system controls the operation of the exhaust valves, the processes of starting and reversing the engine, cylinder lubrication, and speed control.

To inject fuel into the working cylinders, fuel pumps are used, which are essentially hydraulic pressure multipliers. Structurally, they are simpler than their mechanical counterparts. Thus, the use of hydraulics made it possible to abandon the camshaft with its drive, the mechanisms for adjusting the cyclic supply and controlling the fuel injection phases (Variable Injection Timing, further – VIT). The plunger has no control edges, which significantly simplifies its manufacturing technology and increases its service life. In addition, the elements used to supply and supply fuel to the working cylinders under high pressure do not require any changes either in design or in terms of maintenance. The same fuel injectors, high-pressure pipelines, and connecting fittings are used here as in traditional injection systems.

The fuel pump consists of a hydraulic cylinder and a plunger sleeve. The hydraulic cylinder houses a piston, which is mechanically connected to the plunger of the fuel pump (Fig. 5.47).

The diameter of the hydraulic drive cylinder is several times larger than that of the plunger. This allows the fuel pressure at the pump outlet to be increased by the same amount, raising it from 20 MPa for ME series engines and 30 MPa for UEC Eco-Engine engines to 60...100 MPa. Moreover, unlike a mechanically driven fuel injection pump, the pressure under which fuel is supplied to the injector does not depend on the engine speed, which significantly improves the quality of its combustion at partial loads.

For preliminary review

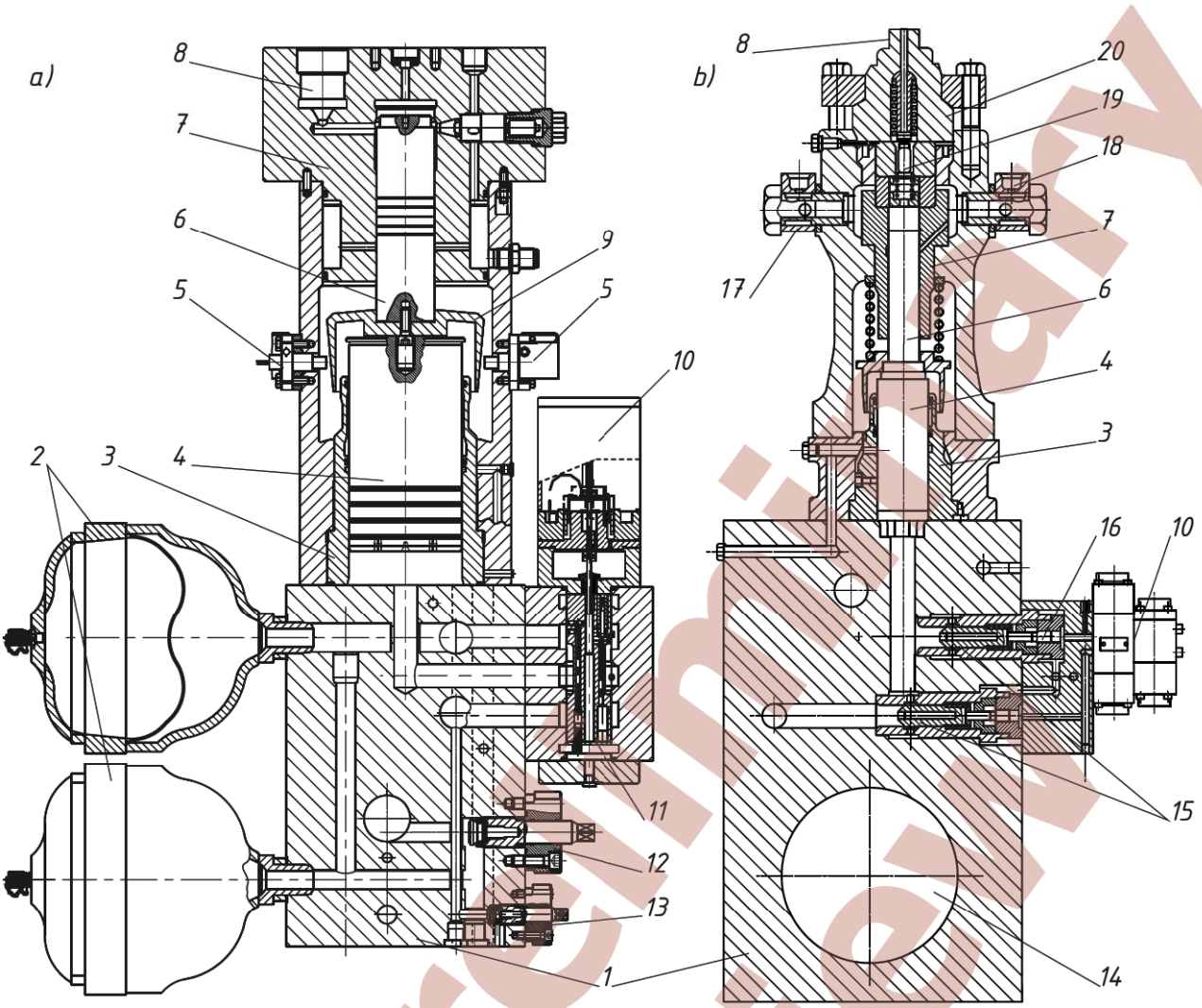


Figure 5.47 – Fuel pumps with hydraulic drive of low-speed diesel engines of the ME series from MAN (*a*) and the UEC Eco-Engine series from Mitsubishi (*b*): 1 – injection pump housing; 2 – gas pressure accumulators; 3 – hydraulic piston bushing; 4 – hydraulic piston of the injection pump plunger drive; 5 – displacement sensors; 6 – plunger; 7 – plunger bushing; 8 – connecting element of the high pressure line; 9 – measuring cone; 10 – solenoid; 11 – control spool; 12 – control pressure adjustment valve; 13 – control pressure limitation valve; 14 – accumulator cavity; 15 – control valves; 16 – hydraulic piston of the control valve drive; 17 – fuel supply fitting; 18 – fuel outlet fitting; 19 – delivery valve; 20 – fuel injection pump cover (adapted from [36, 37])

For Mitsubishi UEC Eco-Engine series engines, the role of a pressure accumulator is played by a cavity in the rigid housing of the pump block. This solution made it possible to use a higher pressure to drive the injection pump and exhaust valve and made it possible to reduce the amount of oil in the system.

For MAN ME series engines, each pump block is equipped with gas pressure accumulators with a diaphragm separator, which ensure a stable oil supply and eliminate the occurrence of dangerous fluctuations in the control oil system. Oil flows to the diaphragm pressure accumulators from the control line and from them to the distribution devices of the fuel pump, exhaust valve actuator, and lubricator pump.

The fuel pumps of the UEC Eco-Engine series engines use two-stage electrically controlled valve-type distributors. The hydraulic diagram of the fuel pump of this diesel engine is shown in Fig. 5.48 *a*.

The ME series diesel engines use a two-stage electrically controlled spool-type distribution mechanism, the diagram of which is shown in Fig. 5.48 *b*.

For preliminary review

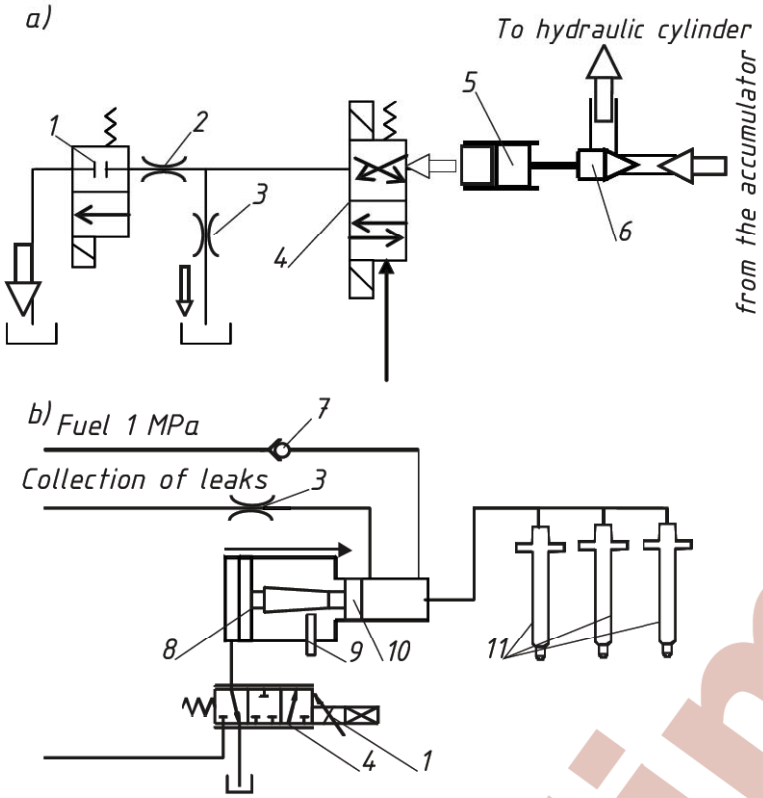


Figure 5.48 – Schematic diagrams of the hydraulic drive of the injection pump of the UEC Eco-Engine series engine from Mitsubishi (*a*) and ME from MAN (*b*): 1 – pilot valve driven by a solenoid; 2 – main throttle; 3 – drain throttle; 4 – main distributor; 5 – main valve drive piston; 6 – main valve; 7 – check valve; 8 – hydraulic piston; 9 – position sensor; 10 – fuel plunger; 11 – injectors (adapted from [9, 37])

The distribution mechanisms of fuel pumps and exhaust valve actuators, as a rule, have an identical design, and in some cases they are unified. The operating principle and general design of spool valves for third generation ME series engines are shown in Fig. 5.49.

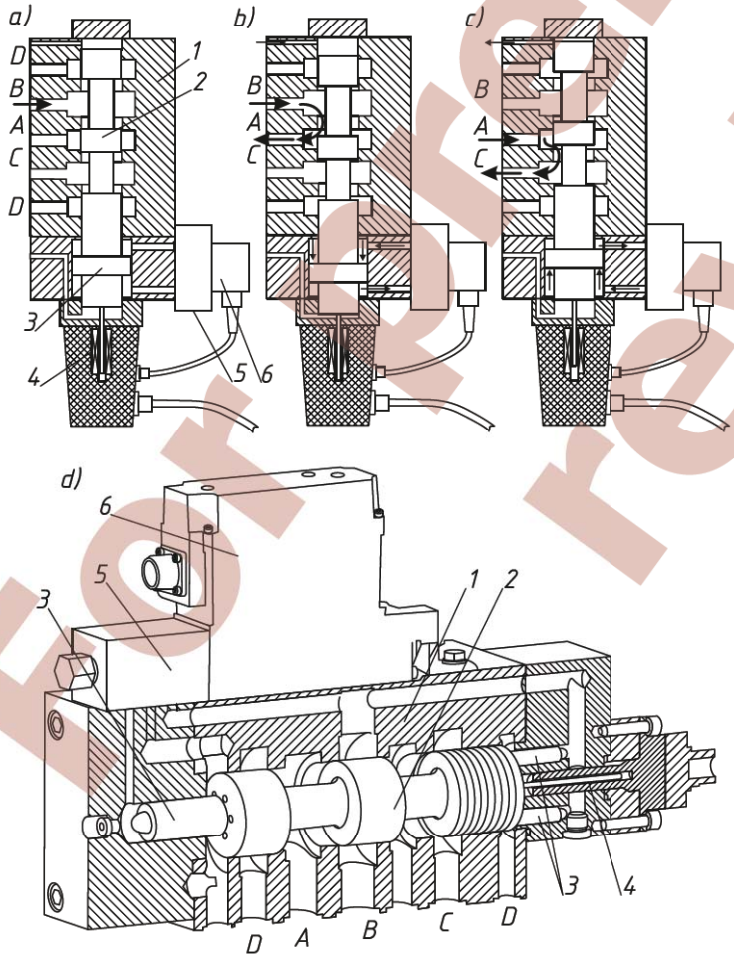


Figure 5.49 – Diagram of the operation of the spool distributor of the injection pump of the ME series engine from MAN (*a*, *b*, *c*) and its general view (*d*): 1 – housing; 2 – main spool; 3 – main spool drive piston; 4 – displacement sensor; 5 – pilot valve; 6 – solenoid; *A* – hydraulic cylinder cavity; *B* – high pressure cavity; *C* – drain cavity; *D* – leakage collection cavity (adapted from [38])

For preliminary review

The main spool with cut distribution grooves is located in the distributor housing.

The main spool moves using a protrusion that acts as a hydraulic piston. Oil enters the cavities located on both sides of the protrusion through the so-called «pilot» spool valve with an electrical drive.

If there is no signal, the spool takes the middle position, with its central edge blocking the communication between the cavity of the drive hydraulic cylinder *A* and the high-pressure cavity *B* and the drain cavity *C*. The plunger remains motionless, there is no injection (Fig. 5.49 *a*).

When an electrical signal is received from the control unit, oil is supplied to the cavity above the control protrusion, moving it and the entire spool down. As a result, the cavity of the hydraulic piston *A* connects with the high-pressure cavity *B*, and the piston together with the plunger begins to move upward (Fig. 5.49 *b*). When the opening pressure of the injector in the cylinder is reached, fuel injection begins. The magnitude of the cyclic supply depends on the stroke of the plunger during the discharge stroke, which is determined by the duration of the opening of the spool valve.

At the end of the discharge stroke, the control valve supplies oil to the cavity under the projection, the spool moves upward, connecting cavity *A* with drain cavity *B* (Fig. 5.49 *b*). The oil clears the cavity of the hydraulic cylinder, the injection stops, and the plunger returns to its original position.

To collect leaks in the spool pair, the distributor housing is equipped with additional drainage cavities *D*. Control of the position of the spool and its movement is carried out using a feedback sensor, which generates and transmits a signal to the electronic control unit. The design of switchgears on different engine models may differ. So, in Fig. 5.49 *d* shows a general view of the distributor, in which the pistons for driving the main spool are located at its ends.

The moment of oil supply under the drive piston, as well as the beginning of its draining, is determined by the receipt of a control pulse from the electronic unit, which is equipped with each engine cylinder.

An electronic control system based on microprocessor technology in combination with a hydrostatic injection pump drive system allows for the implementation of various methods of supplying fuel to the cylinders, single injection of the entire cyclic portion, preliminary injection of the so-called pilot portion of fuel preceding the main injection, multi-stage injection, etc. The corresponding fuel supply law can be optimized for a specific engine operating mode and can be applied when arriving at this mode.

The transition from one fuel supply control algorithm to another can be performed in the time interval between two successive injections.

To ensure the normal functioning of the hydrostatic drive of the injection pump, exhaust valve and lubricator pumps, the engines are equipped with a control oil system. A schematic diagram of the MAN ME series engine system is shown in Fig. 5.50.

The oil in the system is taken from the engine's circulating lubrication system. Before entering the control line, it undergoes additional cleaning in a self-cleaning filter, which retains mechanical impurities up to 5 μm in size. Next, the oil is supplied to the pump block. Some of the pumps are driven by the engine and supply oil to the system at a pressure of 20 MPa during diesel operation.

Some of the pumps are electrically driven and provide the system with oil at a pressure of 17.5 MPa during engine start-up or during operation in emergency situations. After the pumps, the oil enters the accumulating cavity, which reduces pressure pulsations. Next, oil is supplied through high-pressure pipelines to each engine pump unit.

To prevent injuries when oil lines break, they are made in two layers. The internal cavity is used as a working cavity, which is covered on the outside with a protective casing (Fig. 5.51).

To supply oil to the control line, axial plunger pumps are used with capacity control by changing the inclination of the drive disk.

The placement of pumps on the engine drive box, the pump operation diagram and its structure are shown in Fig. 5.52.

The stroke of the plungers in pumps of this type depends on the angle of inclination of the drive disk to the axis of the cylinder block. Changing the angle of inclination of the disk through a hy-

For preliminary review

draulic mechanism allows you to adjust the pump performance from zero flow (the drive disk is perpendicular to the axis of the drive shaft) to maximum (the drive disk is deflected to the maximum angle). This allows, regardless of engine speed, to keep the oil supply to the control line constant, maintaining a given pressure in the system.

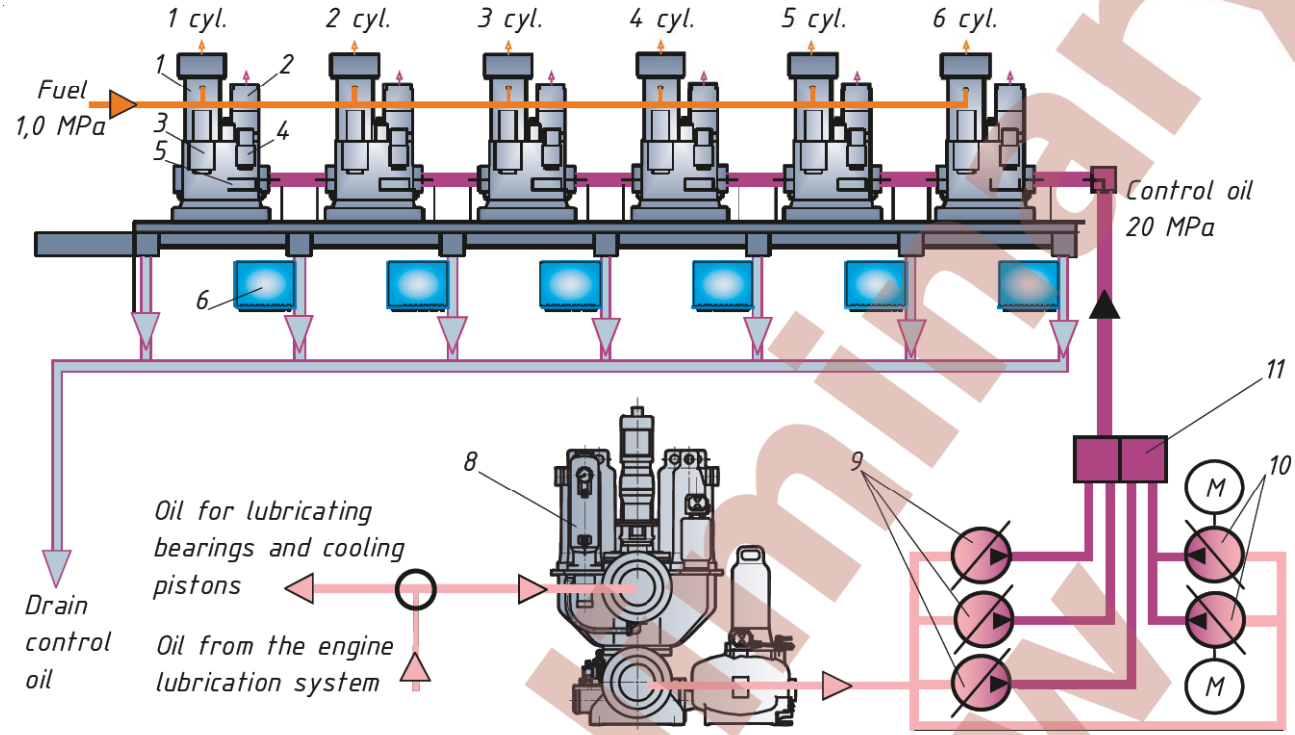


Figure 5.50 – Diagram of the control oil system for the hydraulic drive of the injection pump, exhaust valve actuators and pump-lubricators of low-speed diesel engines of the ME series from MAN: 1 – injection pump; 2 – exhaust valve actuator; 3 – fuel injection pump hydraulic drive control valve; 4 – control valve for the hydraulic drive of the exhaust valve actuator; 5 – lubricator pump; 6 – electronic control unit; 7 – oil distributor; 8 – automatic fine oil filter; 9 – engine-driven oil pumps; 10 – oil pumps driven by an electric motor; 11 – accumulator (adapted from [39])

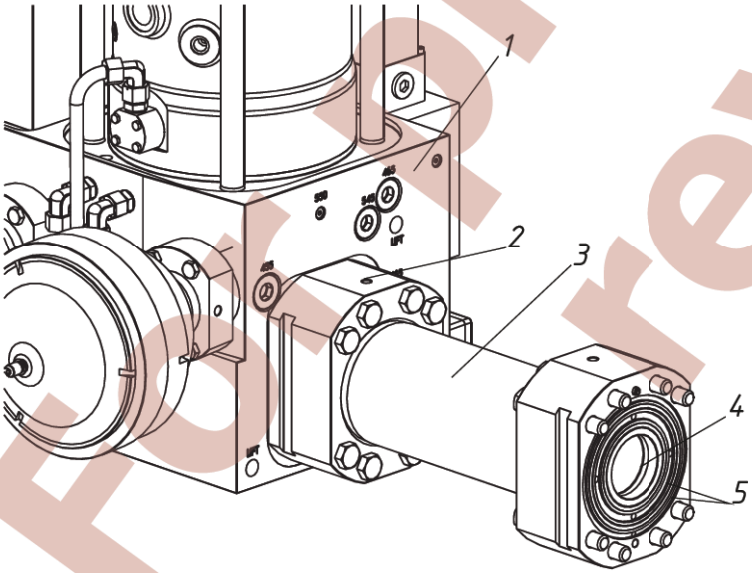


Figure 5.51 – Fragment of the control oil system line: 1 – injection pump housing; 2 – connecting flange; 3 – protective casing; 4 – working surface; 5 – seals (adapted from [20])

The advantages of this type of pump include the absence of valve mechanisms. The liquid sucked in or displaced by the plungers is distributed through crescent-shaped ports in the support disk and holes in the cylinder block. At the dead points of the plungers, the holes of each cylinder are blocked by the lower and upper dividing bridges located between the ports. As a result, such pumps have a fairly high efficiency with a relatively small supply unevenness, and when the direction of rotation of the engine changes, they keep the direction of pumping the liquid unchanged.

For preliminary review

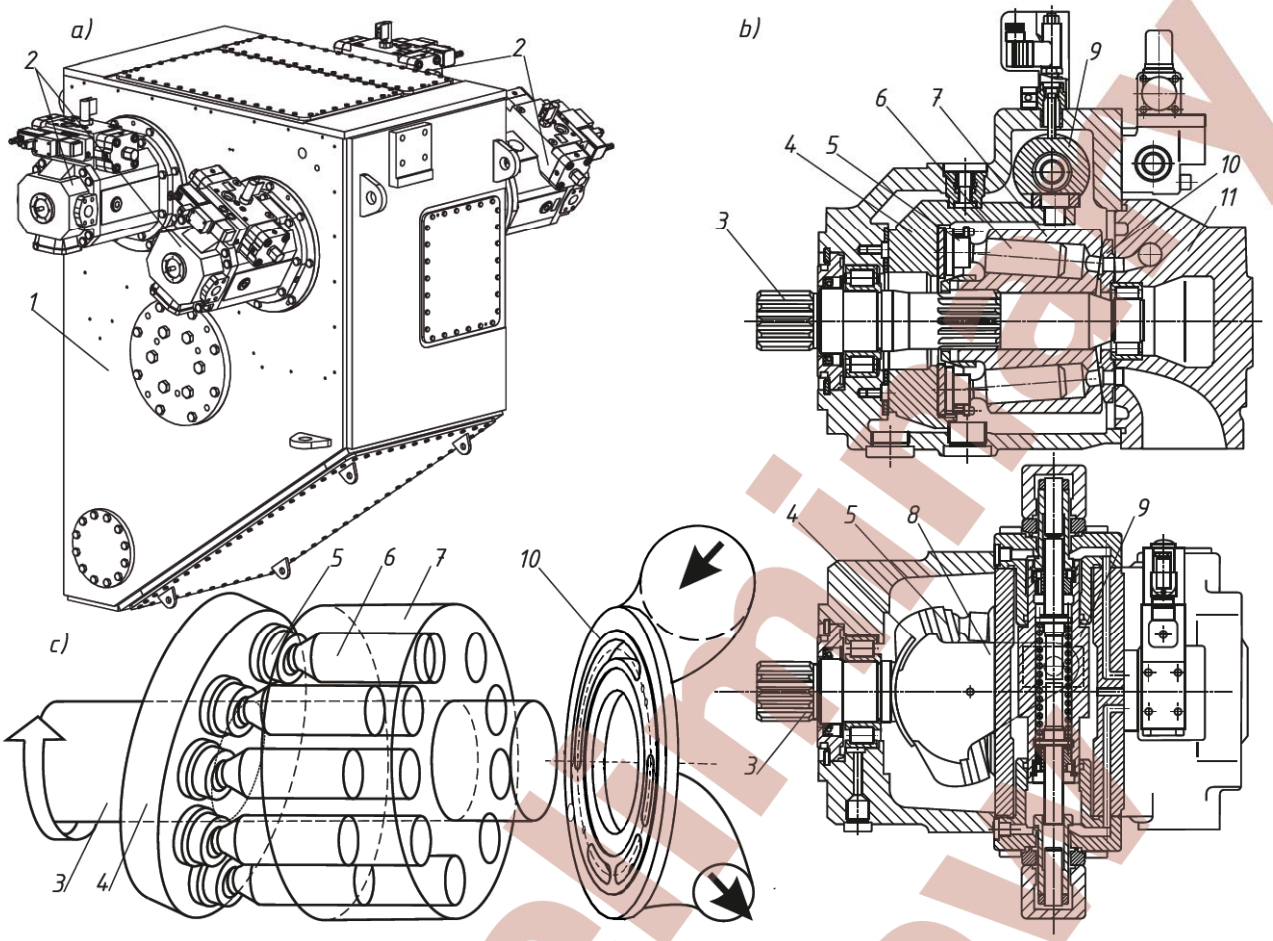


Figure 5.52 – Axial plunger type oil pumps, control hydraulic drive systems for low-speed diesel engines of the ME series from MAN: *a* – location of the pumps on the engine drive box; *b* – operating diagram of an axial plunger pump; *c* – pump device; 1 – drive box; 2 – oil pumps; 3 – pump drive shaft; 4 – inclined drive disk; 5 – ball bearing support; 6 – plunger; 7 – stator; 8 – lever for changing the tilt of the drive disk; 9 – mechanism for changing the inclination of the drive disk; 10 – support-distribution disk; 11 – cover (adapted from [40, 41])

5.6.2 Hydraulically driven pump-injectors for MAN low-speed engines

The constant tightening of environmental standards in shipping has required manufacturers to develop new, more environmentally friendly types of fuels, and with them new approaches to equipment for supplying them to the working cylinders of the engine. The choice of new approaches was also influenced by the fact that the engines had to retain the ability to operate on traditional fuels. As alternative fuels, not only gas fuels were considered, the use of which will be discussed in a separate chapter, but also conventionally liquid fuels, such as methanol, ethanol, ammonia and liquid volatile organic compounds. For large engines with a working cylinder diameter 50 cm and above dimensions S and G, MAN has developed a new concept for fuel equipment and exhaust valve drive mechanism, designated Mk 10. The main components of the new operating processe control system are elements such as an exhaust valve with top control (Top Controlled Exhaust Valve – TCEV) and a hydraulically driven pump-injector (Fuel Booster Injection Valve – FBIV). For the version of engines capable of running on methanol, FBIV has been modified, since the use of methanol requires additional cooling and sealing oil to prevent it from entering the hydraulic drive system. Installing new TCEV and FBIV components on the engine is an alternative to the traditional hydraulic cylinder block for previous generations of ME engines. As a result of design optimization, it was possible to reduce the weight of the engines by approximately 30%. The design and principle of operation of the TCEV module will be discussed in the corresponding chapter.

For preliminary review

Hydraulic pump-injector (FBIV module)

At the design stage of a new injection system, the company's specialists considered two design options for the FBIV module. The first option (Version *A*) was equipped with a module with direct control from an electronic unit, which is a control spool valve with an electrohydraulic drive and a feedback sensor (Fig. 5.53 *a*). The second option is a simplified version without a separate control valve on each FBIV module (Version *B*) (Fig. 5.53 *b*).

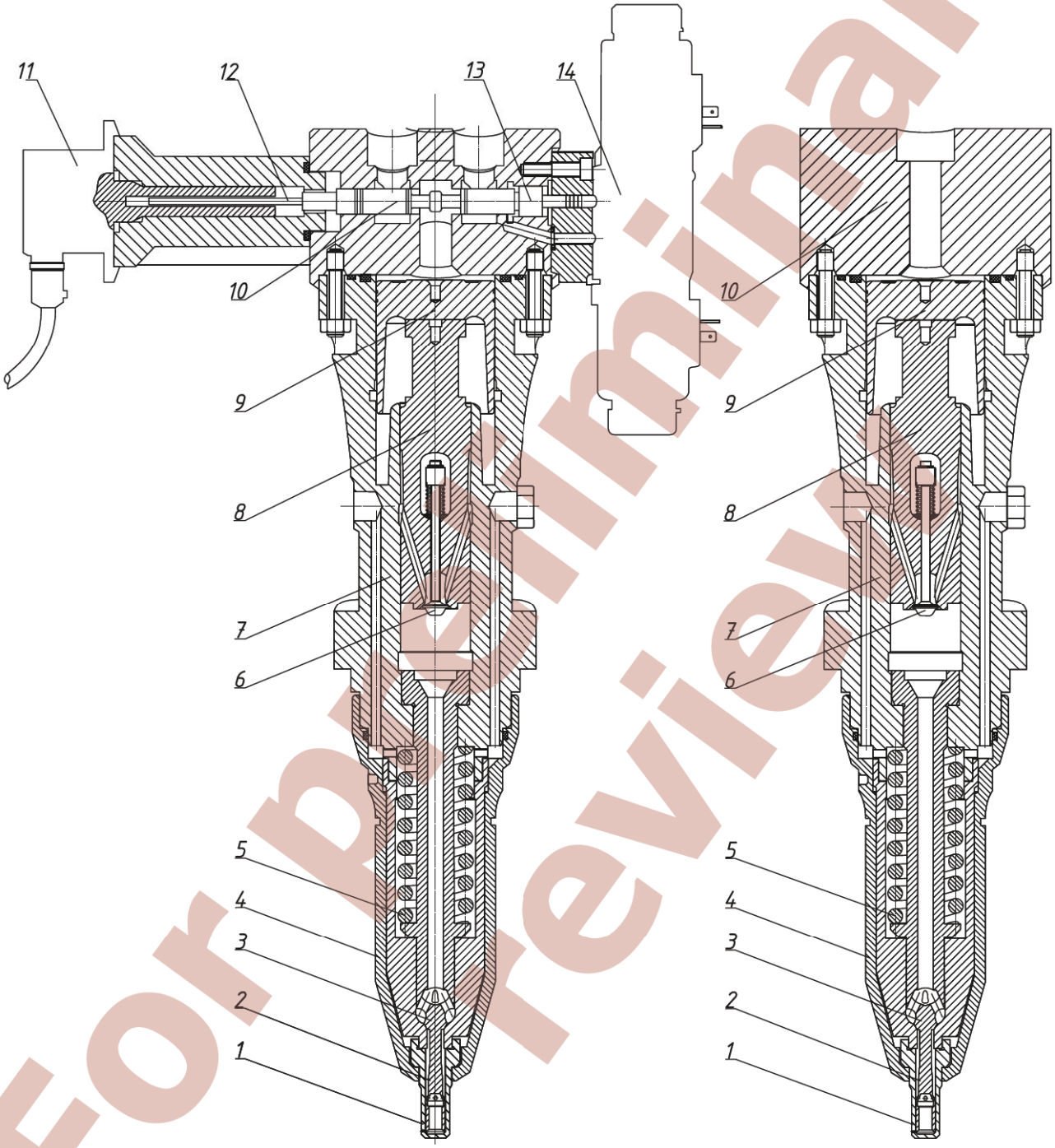


Figure 5.53 – Hydraulic pump-injectors of low-speed engines from MAN series Mk 10 version *A* (*a*) and in version and *B* (*b*): 1 – spray tip; 2 – spool valve; 3 – needle valve; 4 – cap; 5 – needle valve spring; 6 – suction valve; 7 – pump-injector housing; 8 – plunger; 9 – hydraulic piston; 10 – control spool; 11 – control spool position sensor; 12 – measuring rod; 13 – hydraulic piston of the control spool drive; 14 – solenoid control valve (adapted from [42, 43])

The operating procedure of the pump-injector version *A* is shown in Fig. 5.54.

For preliminary review

During the period between fuel injections, the sub-plunger space is filled through a suction valve mounted directly into the plunger part. The valve opens automatically under the influence of pressure maintained in the pressure line (Fig. 5.54, *a*).

When an electrical signal is applied to the electronic control valve, control oil enters the spool valve drive cavity located on the right. As a result of this, the spool moves to the position corresponding to the supply of oil to the hydraulic cylinder of the plunger drive. The hydraulic piston begins to move downward, moving the plunger. Given the difference in the diameters of the hydraulic piston and plunger, the fuel pressure in the sub-plunger cavity increases significantly, which leads to the closure of the suction valve. A further increase in pressure causes the needle valve to overcome the force of the spring loading it and opens. Fuel is injected into the engine combustion chamber (Fig. 5.54, *b*). Duration supply and the amount of fuel supplied by the pump-injector depends on the opening time of the control spool valve.

To end the fuel supply, a corresponding signal is sent to the control valve, which leads to the supply of control oil to the right cavity of the spool valve drive. As a result, the valve moves to the left and connects the cavity of the hydraulic cylinder with the drain line. Under the influence of fuel pressure in the filling line, the plunger rises upward, moving the hydraulic drive piston to its original position (Fig. 5.54 *c*).

A metal rod connected to the slide valve moves inside the induction sensor. The change in inductance is transformed into an electrical signal, through which the electronic control system controls the position of the valve.

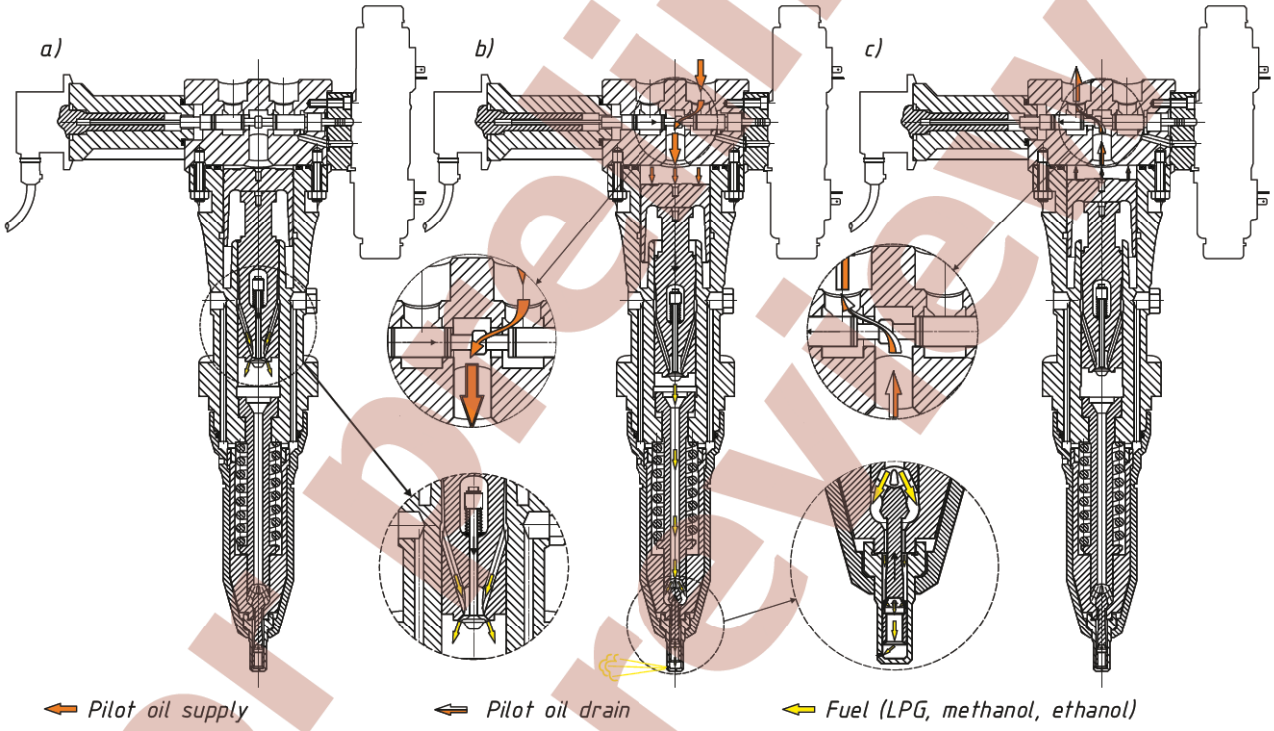


Figure 5.54 – The operating procedure of the pump-injector of low-speed MAN engines of the Mk 10 series with a hydraulic drive: *a* – initial state, filling the sub-plunger space with fuel; *b* – oil supply to the hydraulic cylinder, fuel injection; *c* – draining the oil from the hydraulic cylinder, returning to its original state (adapted from [43])

As already noted, the second version of hydraulic pump-injectors represents a simplified version without a separate control valve on each FBIV module (Fig. 5.53 *b*). Control oil is supplied to all modules installed on the cylinder cover from a common line. The oil supply is controlled by one electro-hydraulic valve installed on the valve assembly housing.

Tests carried out for both versions showed that, despite the slower dynamic response, the performance of the engine equipped with pump-injectors, version *B*, was not much different from the

For preliminary review

more complex version *A*. For this reason, it was decided to continue work on option *B* for commercial use. It was this configuration that formed the basis of the Mk 10 engine concept. Subsequently, the system proved itself well, providing stable fuel injection from cycle to cycle over the entire range of operating conditions. It was noted that pressure deviations in the cylinders increased with load, but did not exceed the permissible 0.5 MPa at maximum power.

5.6.3 Pump-injectors with a hydraulic plunger drive and electromagnetic control of the fuel supply of high-speed Caterpillar engines

At the turn of the 90s of the last century, Caterpillar and a number of other manufacturers began developing a fundamentally new fuel supply system. It was decided to replace the mechanical drive of the pump-injectors with a hydraulic one, and control the fuel supply using control valves with an electromagnetic drive. In the literature, such a system is called HEUI (Hydraulically actuated Electronic Unit Injection).

The general diagram of the fuel system used for high-speed diesel engines is shown in Fig. 5.55.

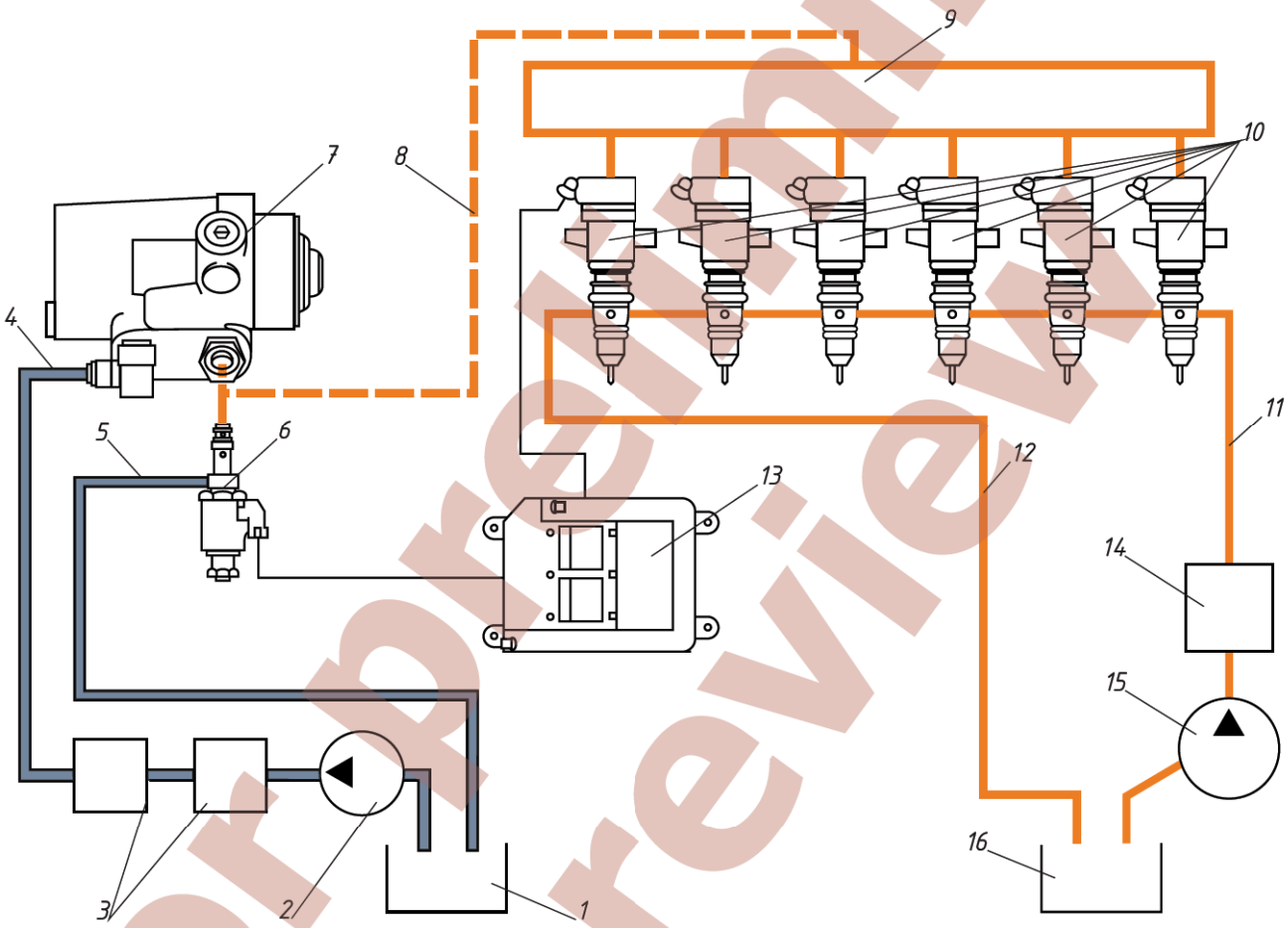


Figure 5.55 – Diagram of a fuel system using hydraulically driven electrically controlled pump-injectors (HEUI): 1 – oil tank for hydraulic drive; 2 – booster oil pump; 3 – oil filters; 4 – oil supply line to the high pressure pump; 5 – oil drain line; 6 – pressure control valve in the hydraulic drive system; 7 – high-pressure oil pump of axial plunger type; 8 – high pressure oil line; 9 – pressure accumulator; 10 – pump-injectors with hydraulic drive; 11 – fuel supply line; 12 – fuel outlet line; 13 – electronic control unit; 14 – fuel filter; 15 – fuel priming pump; 16 – fuel storage tank (adapted from [44])

The system has two independent circuits – fuel and oil.

Along the fuel circuit, purified fuel is supplied, using a gear pump under a pressure of 0.2...0.55 MPa, to the pump-injectors through channels made, as a rule, in the form of drillings in the cylinder head. The remaining unused fuel is drained back into the supply tank through the throttle valve.

For preliminary review

The oil circuit consists of low and high pressure lines. The low pressure line serves to clean, cool and supply control oil from the storage tank to the high pressure oil pump. The high-pressure circuit consists of supply pipelines, an oil pressure regulator, various control sensors and a pressure accumulator, with which the pump-injectors communicate using connecting pipelines. The pressure under which oil is usually supplied to the pump-injectors is in the range of 27...30 MPa. The general arrangement of the HEUI system pump-injector, oil accumulator and connecting pipeline is shown in Fig. 5.56.

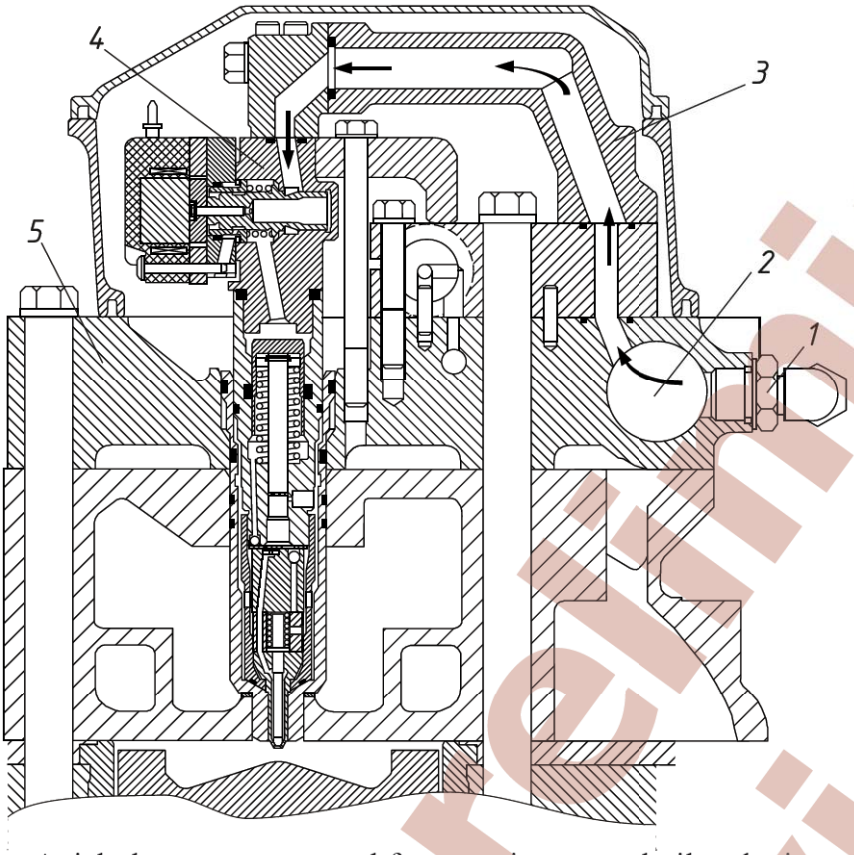


Figure 5.56 – General arrangement of an electrically controlled hydraulically driven pump-injector, oil accumulator and connecting pipeline on a Caterpillar 3400 series engine: 1 – supply line; 2 – oil pressure accumulator; 3 – connecting pipeline; 4 – pump-injector with hydraulic drive; 5 – cylinder head (adapted from [45])

Axial plunger pumps used for pumping control oil make it possible, by changing the supply, to maintain the pressure in the control line at a given level even at minimum engine speeds. As a result, the injection pressure by pump-injectors practically does not depend on the rotation speed of the diesel crankshaft, and the ability to regulate the pressure in the control oil circuit allows you to change the injection pressure in a wide range of load-speed modes of engine operation (Fig. 5.57).

The first generation pump-injectors are units that include a nozzle with a needle valve and a hydraulic pressure multiplier consisting of a hydraulic piston that actuates the fuel supply plunger (Fig. 5.58 *a*, *b*).

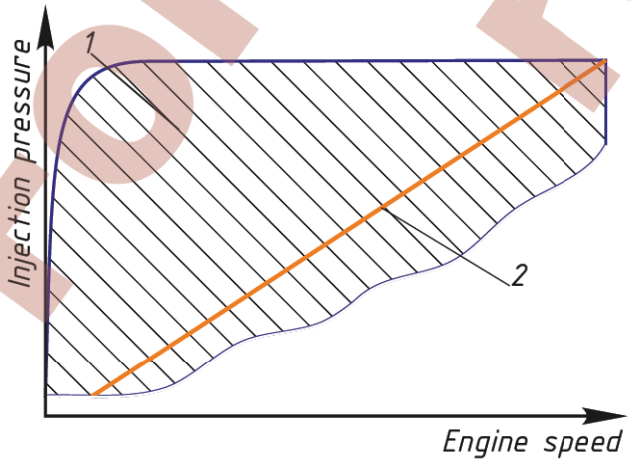


Figure 5.57 – Dependence of injection pressure on rotation speed: 1 – for HEUI type fuel systems; 2 – for standard mechanically driven pump-injectors (adapted from [44])

Oil is supplied to the cavity of the hydraulic cylinder through a poppet valve, which can have a

For preliminary review

horizontal (Fig. 5.58 *a*) or vertical (Fig. 5.58 *b*) arrangement. The valves are opened by a solenoid switched from the electronic control unit, and closed by a return spring. The valve is pressed against the seat under the influence of pressure in the oil line. In the open state, the valve communicates the high-pressure cavity with the cavity of the hydraulic cylinder, and in the closed state, with its unloading device, it communicates with the cavity of the hydraulic cylinder with the drain line. The unloading device can be made in the form of a cylindrical spool (Fig. 5.58 *a*) or a shut-off cone (Fig. 5.58 *b*).

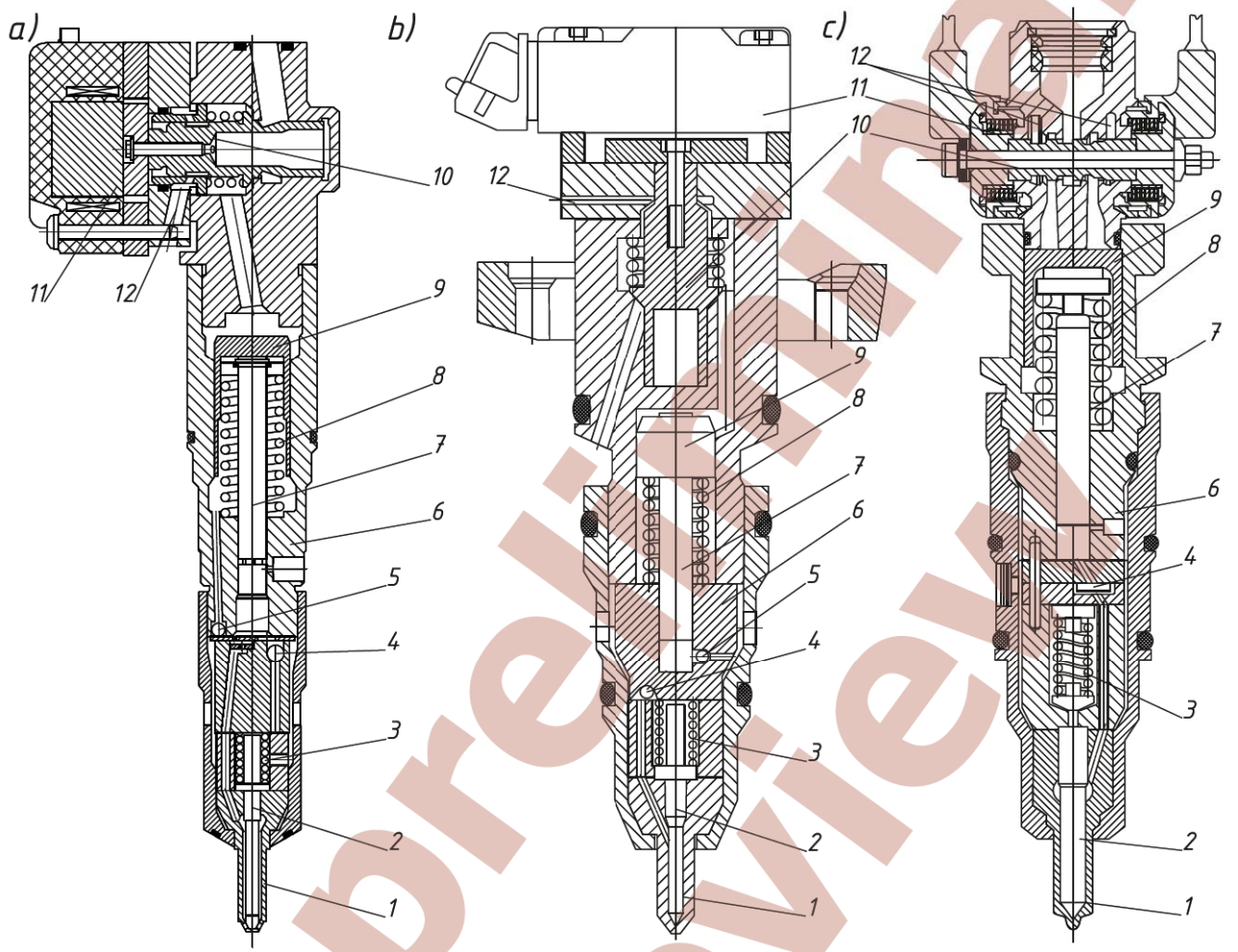


Figure 5.58 – Pump-injectors with hydraulic drive and electromagnetic supply control (HEUI): *a* – first generation with a side-mounted control valve; *b* – first generation with a central location of the control valve; *c* – second generation G2 with double bistable control valve: 1 – spray housing; 2 – needle valve of the nozzle; 3 – needle valve spring; 4 – suction valve; 5 – check valve; 6 – plunger bushing; 7 – plunger; 8 – return spring; 9 – hydraulic piston of the plunger drive; 10 – control valve; 11 – solenoid of the electromagnetic drive of the control valve; 12 – drain line (adapted from [44, 45, 46])

Experience in operating these pump-injectors has shown that their use is only possible on relatively small engines with limited cyclic supply volumes. To ensure the necessary speed when increasing cyclic supplys, it became necessary to increase the size of the control valve, which would lead to the need to apply significant forces to open it. Creating the required force using a solenoid turned out to be quite problematic, since it would require large currents through the coil, which, given the limited size of the solenoid, would lead to its rapid overheating and failure.

Therefore, in their further developments, manufacturing companies switched to the use of bistable spool valves.

The second generation pump-injector type G2 with a double bistable control valve is shown in Fig. 5.58 *c*.

Unlike the first generation pump-injectors, the opening and closing of the spool valve here is car-

For preliminary review

ried out using two solenoids. In this case, all axial forces acting on the spool balance each other, so the spool can be in any position, maintaining it even after the moving force on it ceases (Fig. 5.59). That is, the spool can be in the position of supplying oil to the hydraulic cylinder for any length of time or in the position corresponding to draining the oil, without additional effort to hold it in these positions.

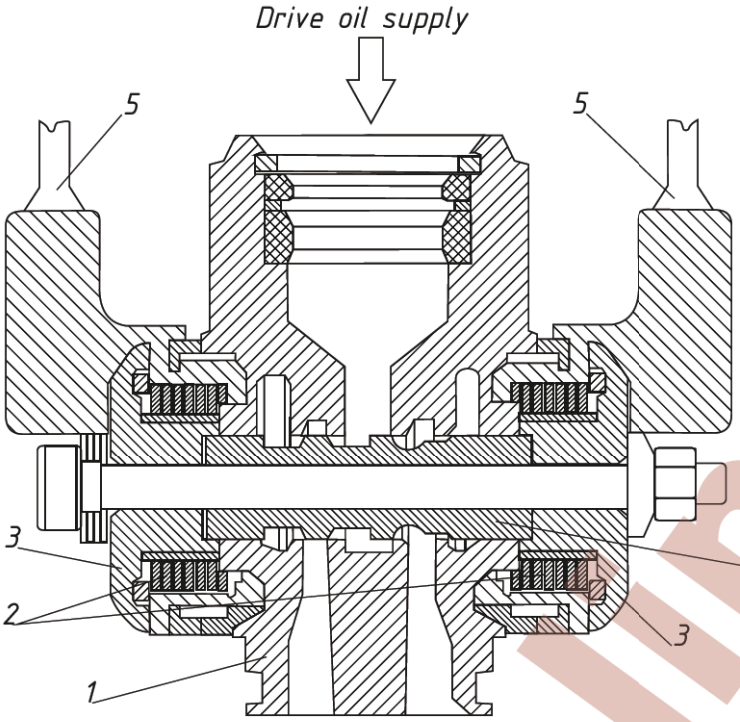


Figure 5.59 – Bistable spool valve for supply control: 1 – valve housing; 2 – solenoid coil; 3 – solenoid covers; 4 – spool; 5 – terminal for connecting to the electronic control unit (adapted from [44])

This provides a number of significant advantages in organizing fuel supply.

Firstly, moving a spool balanced by axial forces requires much less force than opening a valve pressed by oil pressure and a return spring. This makes it possible to reduce the size of the solenoid and increase the valve response speed. For the same purposes, the spool valve is made double, which allows the flow area to be doubled and the spool stroke to be reduced.

Secondly, a short pulse is enough to move the spool from one position to another, after which the valve itself maintains its new position. Thus, the time it takes for the current to pass through the coil is minimized, and the magnitude of the currents during the control pulse can reach fairly high values without damaging the solenoid winding.

Regardless of the type, all pump-injectors shown in Fig. 5.58, after the control valve opens, oil from the control line enters the cavity of the hydraulic cylinder, causing the piston to move down. In this case, the plunger, rigidly connected to the piston, begins to displace fuel from the sub-plunger cavity through the check valve into the fuel injector. Considering that the area of the plunger is several times smaller than the area of the hydraulic piston, the pressure in the sub-plunger cavity increases to 170...210 MPa.

As the pressure in the fuel injector cavity increases, the fuel opens the needle valve, overcoming the compression force of the loading spring, and then, through the nozzle holes of the fuel injector, it is injected into the engine combustion chamber.

To supply fuel and control oil to the pump-injectors, cavities in the cylinder head are used. To separate the individual cavities, grooves are provided along the pump-injector housing for installing O-rings.

For engines with a sufficiently high cylinder power and, therefore, an increased cyclic portion of fuel, pump-injectors with two-stage control of the oil supply to the hydraulic cylinder of the plunger drive were developed. This modification of pump-injectors is called HI 300B. The general structure of a pump-injector of this type is shown in Fig. 5.60. The operating procedure of the pump-injector type HI 300B is shown in Fig. 5.61.

For preliminary review

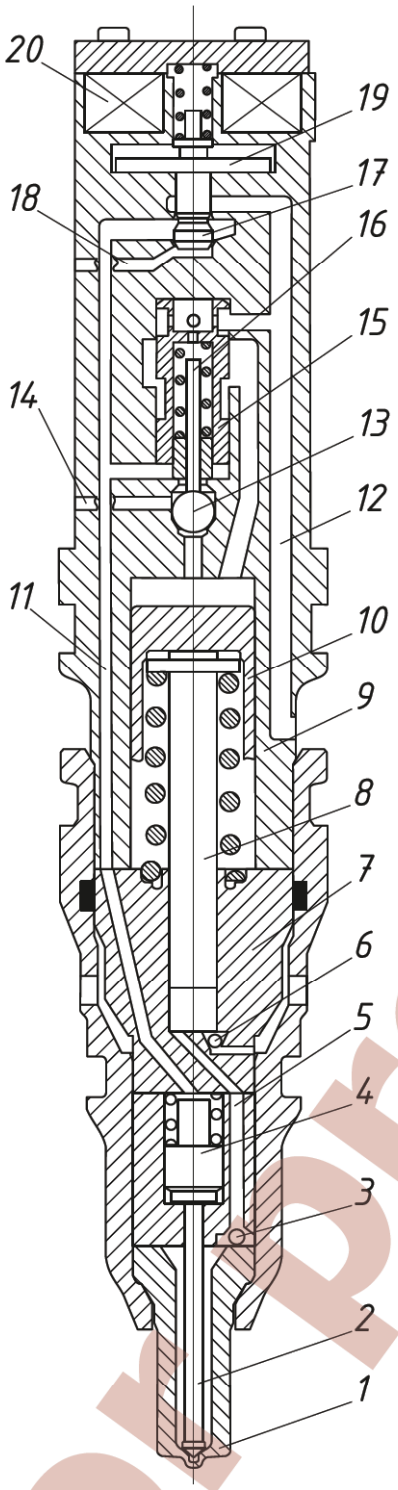


Figure 5.60 – Pump-injector type HI 300B: 1 – spray housing; 2 – needle valve; 3 – check valve; 4 – hydraulic loading piston of the needle valve; 5 – channel for supplying fuel to the nozzle; 6 – suction valve; 7 – plunger bushing; 8 – plunger; 9 – hydraulic cylinder housing; 10 – hydraulic piston; 11 – channel for supplying control oil to hydraulically close the spray valve; 12 – oil supply channel to the control valve; 13 – control oil drain valve; 14 – control oil drain channel; 15 – spool distributor of control oil; 16 – loading rod of the power circuit drain valve; 17 – control valve; 18 – control oil drain channel; 19 – solenoid armature; 20 – solenoid coil (adapted from [44])

A distinctive feature of these devices is that they are equipped with two oil circuits – control and power. For pump-injectors of this type, the electric solenoid valve does not control the supply of oil to the hydraulic cylinder cavity, but drains it from the control line. In addition, the nozzle needle valve is equipped with a hydraulic locking system, which increases the reliability of its operation and makes it possible to ensure a sufficiently high initial pressure of fuel injection into the engine combustion chamber and an abrupt shutdown of the supply.

In the absence of a control signal to the solenoid coil, the poppet-type control valve is held in its lowest position by the force of the return spring and oil pressure, connecting the high-pressure oil line to the control circuit of the pump-injector. The lower flange of the valve blocks the drain line of the control circuit. Oil under a pressure of 26 MPa fills the channels of the control circuit, acts on the loading piston of the needle valve of the nozzle and holds the spool valve of the power circuit oil in the uppermost position (Fig. 5.61 *a*).

For preliminary review

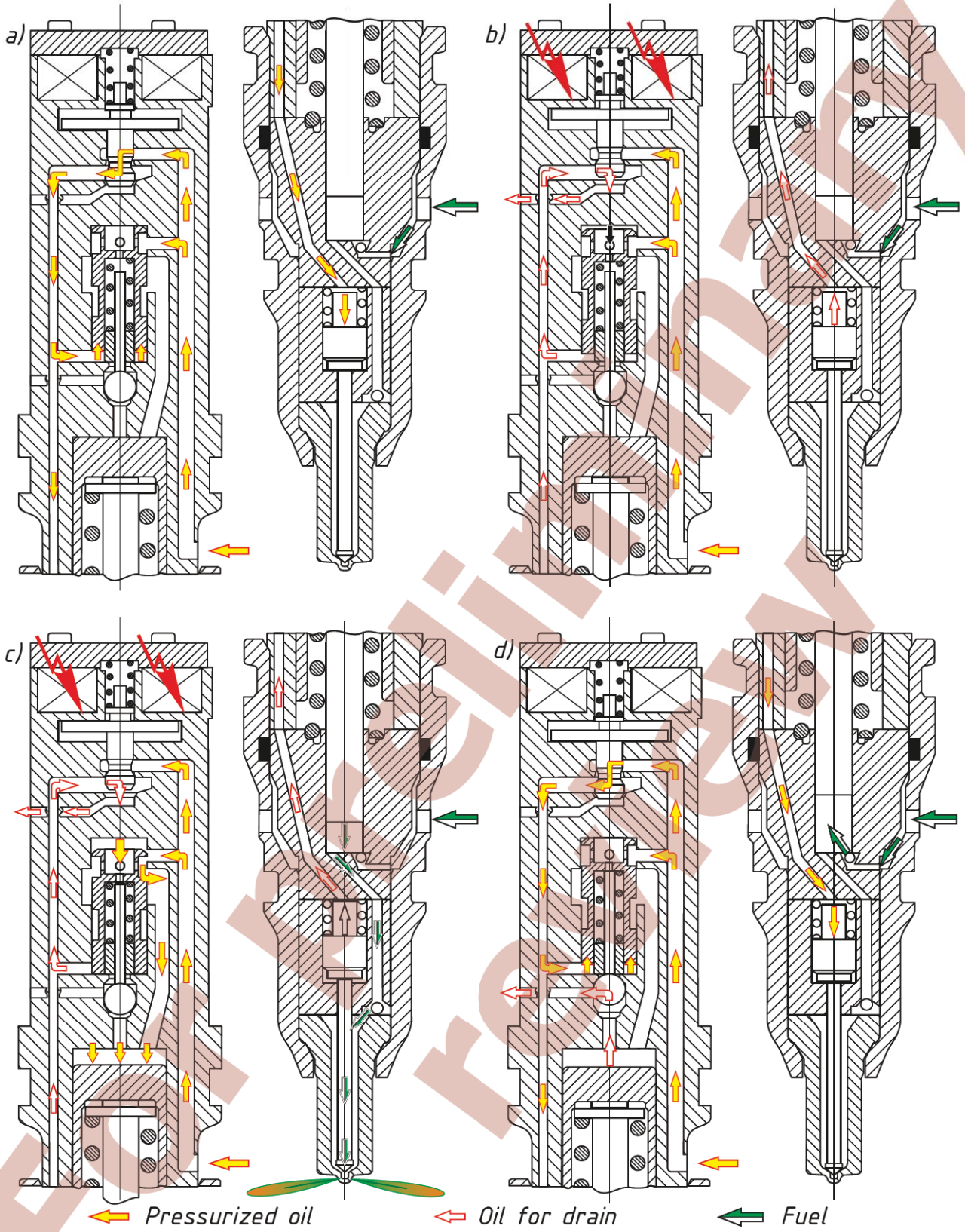


Figure 5.61 – Operating order of a pump-injector type HI 300: *a* – initial state, the needle valve is closed under control oil pressure; *b* – the control valve is open, draining oil from the needle hydraulic locking system, moving the spool valve down; *c* – the spool valve opens the supply of control oil to the hydraulic cylinder, fuel injection occurs; *d* the control valve is closed, the spool valve releases the drain valve stop, the oil is drained into the return line, the sub-plunger space is filled with fuel (adapted from [44])

The control oil spool distributor with its edges blocks the access of oil to the hydraulic cylinder of the plunger drive. In this case, the loading rod of the drain valve is free from the stop and the oil

from the cavity of the hydraulic cylinder freely flows into the drain line of the power circuit. The hydraulic piston, under the action of the return spring, moves to its uppermost position, and a new portion of fuel enters the sub-plunger space through the suction valve.

When voltage is applied from the electronic control unit to the solenoid coil, the control valve moves to its highest position, connecting the cavity of the pump-injector control circuit with the drain line. As a result of the pressure drop in the circuit, the hydraulic locking system of the nozzle needle valve is unloaded.

In this case, the spool valve, under the influence of oil pressure in the power circuit, moves down, connecting the cavity of the hydraulic cylinder with the power circuit (Fig. 5.61 *b*). As it descends, the spool valve acts on the drain valve rod, disconnecting the cavity of the hydraulic cylinder and the drain line.

The oil, acting on the piston, moves it along with the plunger down. Since the plunger area is six times smaller than the hydraulic drive piston area, the pressure in the fuel cavity increases to 170...180 MPa. When the needle valve opening pressure is reached, fuel injection begins into the engine combustion chamber (Fig. 5.61 *c*).

After removing the voltage from the solenoid coil, the control valve moves upward under the action of the return spring, connecting the cavity of the control circuit with the oil line. The pressure in the cavity increases sharply, the oil acts on the loading piston of the needle valve of the nozzle, causing an abrupt stop in flow.

Under the action of the return spring and oil pressure on the lower end of the spool valve, the latter moves upward, disconnecting the cavity of the power circuit and the oil line. In this case, the drain valve rod of the hydraulic cylinder is released, and the oil displaced by the piston flows into the drain line (Fig. 5.61 *c*). The sub-plunger space is filled with a new portion of fuel.

The use of relatively small volumes of oil to drive pump-injectors and two-stage control made it possible to reduce the size of the control valve, which together led to an increase in the speed of the system as a whole. This pump-injector is capable of providing not only standard fuel supply laws, but also some special ones aimed at improving the efficient or environmental characteristics of the engine.

At the same time, the complex hydraulic circuit of pump-injectors does not allow achieving uniform performance in their production. In this regard, the company determines correction characteristics for each unit based on test results, which are supplied with the pump-injector in the form of a software algorithm. This program, in the form of an accompanying file, is installed on the microprocessor control unit when replacing the pump-injector. Subsequently, when generating control signals, the control unit takes into account the individual characteristics of each unit.

Control oil pressure regulator. On small engines, to maintain a given pressure in the control oil system, an electrohydraulic pressure regulator, switched from the electronic control unit, is installed directly in the oil line. The general view of the regulator is shown in Fig. 5.62.

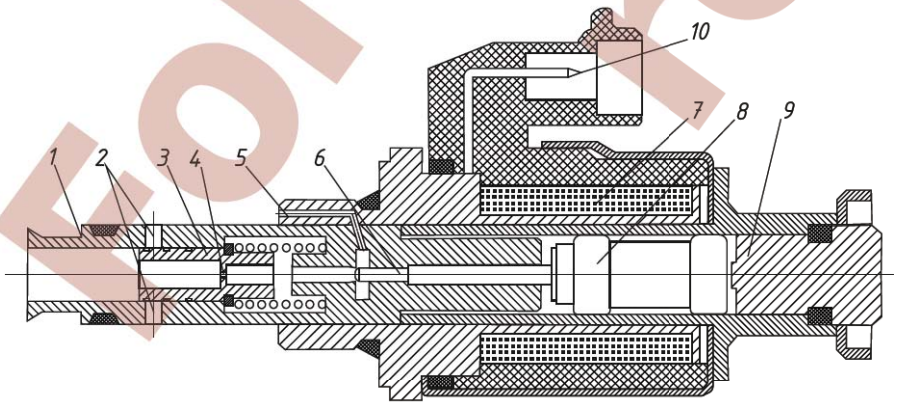


Figure 5.62 – Control oil pressure control valve: 1 – valve housing; 2 – oil drain holes; 3 – spool piston; 4 – throttle hole in the spool piston; 5 – line for draining oil from the differential cavity; 6 – shut-off cone of the system for releasing pressure from the differential cavity; 7 – solenoid coil; 8 – anchor; 9 – anchor stop; 10 – terminal for connecting to the electronic control unit (adapted from [46])

The main regulating housing is the spool piston, which directly interacts with the oil in the high-pressure line. With its edges, the piston blocks the holes for discharging oil into the low pressure

For preliminary review

line. The piston itself has a calibrated drainage channel, through which the high-pressure cavity communicates with the piston space.

In the absence of a control signal to the control solenoid, the pressure relief valve from the piston space is closed, the pressure on both sides of the control piston is the same. Under the action of the return spring, the spool piston moves to its extreme left position, blocking the reset holes.

If the pressure rises above normal, a signal is sent from a pressure sensor installed in the control line to the electronic unit, which generates a control signal to the regulator solenoid. As a result, the drain valve opens and the oil from the piston space is drained into the low pressure line. In this case, the pressure in this cavity decreases sharply, since the throughput of the throttle channel is insufficient to compensate for the oil leakage through the drain valve. Under the influence of the pressure difference, the spool piston moves to the right, opening the drain holes. Some of the oil is drained into the return line, and the pressure in the system decreases. When the drain valve of the piston space is closed, the spool piston returns to its original position.

Control oil pump. In relatively small engines, the pressure in the oil line is usually regulated by bypassing part of the oil to drain. In this case, the performance of the pump depends only on its rotation speed. For medium and high power engines, such regulation leads to unjustified power losses to drive the oil pump. Therefore, in such engines, axial plunger pumps are used to supply oil to the pump-injector drive with performance control by changing the inclination of the drive washer or by changing the magnitude of the discharge stroke of the plungers.

We have already reviewed pumps with the first control method earlier.

In Fig. 5.63 shows the general design of an axial plunger pump with regulation of the supply of control oil by changing the magnitude of the discharge stroke. The main discharge elements are peripherally located around the axis of the plunger, each of which slides along a guide sleeve. Due to the inclined washer, on which the articulated pushers of the plungers and return springs slide, each plunger performs a reciprocating movement when rotating.

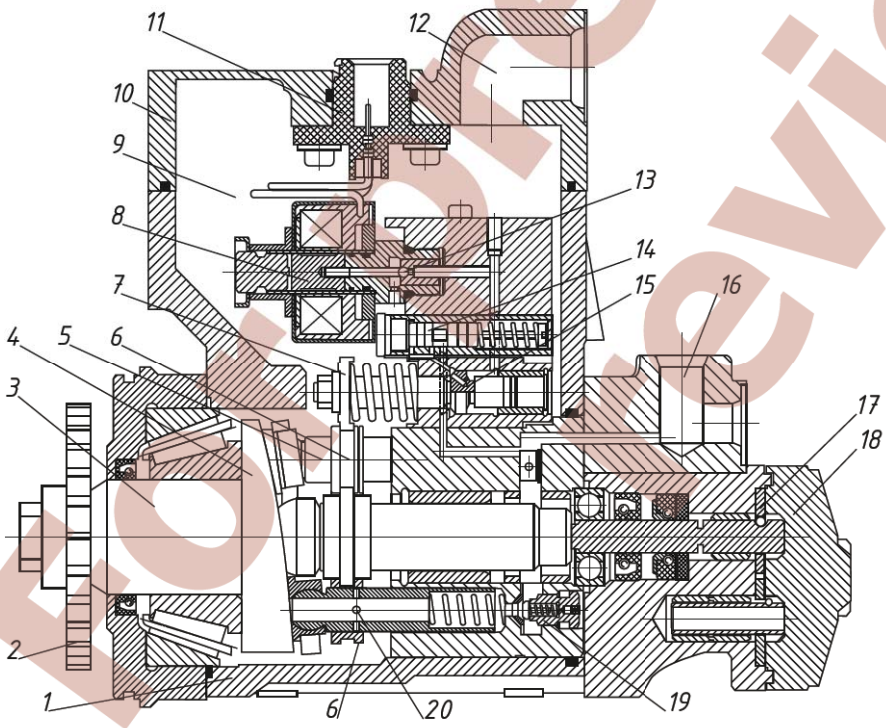


Figure 5.63 – High-pressure axial plunger oil pump of the control oil system with spool valve supply control: 1 – pump housing; 2 – drive coupling half; 3 – drive shaft; 4 – inclined washer of the plunger drive; 5 – plunger; 6 – sliding sleeve for supply control; 7 – sliding bushing drive fork; 8 – solenoid valve for controlling pump performance; 9 – cavity for storing control oil; 10 – oil cavity cover; 11 – electrical connector; 12 – oil inlet from the drain line; 13 – valve for adjusting oil pressure in the supply control system; 14 – spool pressure regulator in the supply control system; 15 – piston for regulating pump performance; 16 – high pressure pressure line; 17 – gear fuel pump; 18 – gear pump cover; 19 – delivery valve; 20 – filling holes (adapted from [47])

At the bottom of each plunger there are filling holes through which oil from the internal space of

For preliminary review

the pump enters the cavity of the plunger.

To regulate the amount of active stroke, a precision-fitted sliding sleeve is located on each plunger.

The bushings of all plungers are connected to each other and to the drive mechanism, which ensures their axial movement, independent of the position of the plunger. At the beginning of the stroke of the plunger, the oil is squeezed back into the cavity of the oil pump, and only from the moment when the filling holes on the plungers are blocked by the sliding sleeve does the discharge stroke of the plunger begin. From this moment, the sub-plunger cavity is closed and the oil is forced into the pressure line through the delivery valve. By changing the position of the sleeve relative to the plunger, you can change the amount of active stroke of the plungers and, consequently, the amount of oil supplied by the pump.

Sufficiently accurate control of the supply depending on the oil consumption at a particular load-speed mode allows you to significantly reduce the power consumption of the oil pump drive.

To change the pump's performance, it is equipped with an automatic control system controlled from an electronic unit. The general structure and operating procedure of this system are presented in Fig. 5.64.

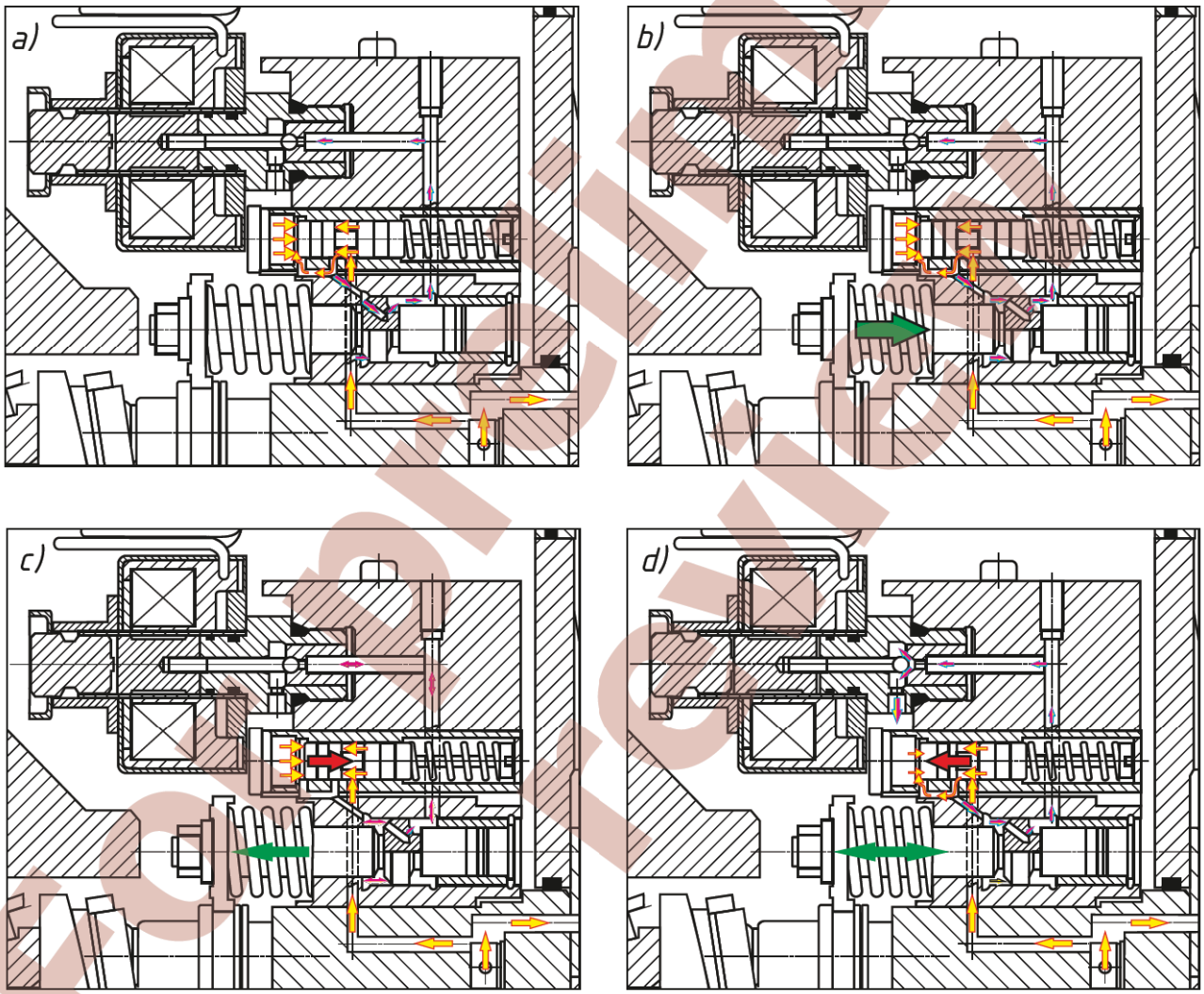


Figure 5.64 – Operating order of the performance regulator of an axial plunger oil pump: *a* – oil is supplied to the pressure regulator in the supply control system; *b* – the pump performance control piston moves to the right, shifting the sliding bushing drive fork in the direction of decreasing flow; *c* – the oil pressure regulator in the supply control system limits the access of oil to the cavity of the piston for regulating pump performance; *d* – the solenoid valve periodically opens, the oil is drained from the control cavity, the pump performance control piston makes a slight translational movement, maintaining the pump performance at a given level (adapted from [47, 48])

Preliminary version

The system consists of two main modules: a module for maintaining constant pressure in the control circuit and a capacity control module.

The pressure regulator in the spool-type supply control system. It is necessary to maintain constant pressure in the control circuit, since the pressure in the discharge cavity can change, and this makes the regulation process more complex.

The regulator consists of a spool piston located in the guide sleeve. On the piston, in its middle part, there is a groove into the cavity of which oil flows from the discharge line. From the same cavity, the oil enters a U-shaped channel connecting the bore cavity and the cavity located at the end of the piston.

Oil is taken from the U-shaped channel into the supply control circuit (Fig. 5.64 *a*). The adjusting piston, under the influence of pressure, begins to move to the right, moving the sliding bushing drive mechanism and reducing pump performance (Fig. 5.64 *b*).

Thanks to the U-shaped channel, oil enters the cavities from both sides of the spool piston under equal pressure. However, due to the difference in area, a force greater in magnitude acts on the end surface of the piston than on its surface from the side of the groove. As a result, the spool piston, overcoming the force of the return spring, moves to the right, blocking the U-shaped channel with the edge on the groove side. The ratio of the areas of the working surfaces of the spool piston and the force of the return spring are selected in such a way that the oil take-off line into the control circuit is blocked at a pressure of 5.7 MPa (Fig. 5.64 *c*). In this way, constant pressure is maintained in the control circuit.

The adjusting piston has a drainage channel through which oil gradually flows from the above-piston cavity to the under-piston cavity, and the drive of the sliding bushings moves in the direction of increasing the flow. If the supply increases more than the value necessary to ensure a given load-speed regime, pressure increases. The pressure sensor sends a signal to the control unit, which, in turn, generates a control pulse to the solenoid of the electrically controlled drain valve. By opening, the valve relieves pressure from the piston cavity of the adjusting piston. The oil flow through the throttle channel in the control piston does not have time to compensate for the loss of oil due to drainage, resulting in a pressure difference, under the influence of which the piston moves in the direction of decreasing flow (Fig. 5.64 *d*). A drop in pressure in the control circuit leads to an imbalance of forces acting on the spool piston, as a result of which it moves to the left, allowing oil from the discharge cavity to enter the supply control circuit. Then the whole cycle repeats. The spool piston, performing reciprocating movements, maintains the pressure in the control circuit at approximately the same level, and the control piston, moving progressively, maintains the specified pump performance.

5.7 Accumulator injection systems

Mechanically controlled accumulator injection systems were somewhat widespread on marine diesel engines in the mid-20th century, but due to high cost, complexity and lack of reliability, they were replaced by systems with direct plunger drives. The current stage of development of new generation accumulator systems began in the 70s of the last century. Their work was based on the principle of electronic control of the fuel supply mode, which ensures the most complete optimization of the diesel operating process over the entire range of load-speed conditions. In literature, such systems are called Common Rail. The widespread introduction of Common Rail systems on marine diesel engines is determined by the need to increase their fuel efficiency and comply with constantly tightening exhaust gas toxicity standards. Experience in recent decades has shown that significant progress in improving efficiency and environmental performance can be achieved by improving the fuel delivery process.

The main criteria for the perfection of the fuel injection process are indicators of efficiency, power, noise, exhaust gas toxicity, as well as compliance with restrictions on cylinder pressure, combustion severity, thermal loads, and gas temperature in front of the turbine. Today, electronically controlled accumulator injection systems make it possible to most fully satisfy the increased requirements for improving operational processes in marine diesel engines in all operating modes.

For preliminary review

From Fig. 5.65 it can be seen that with traditional fuel injection pumps, the injection pressure decreases in proportion to the decrease in the rotational speed of the fuel cam, and decreases significantly with a decrease in the cyclic fuel supply.

In accumulator systems, injection pressure can be maintained constant regardless of these factors. Moreover, injection pressure and fuel delivery phases are controlled independently of each other. Fuel injection begins at maximum pressure, which contributes to high-quality atomization of the first portions of fuel entering the combustion chamber and faster pre-flame processes.

The use of electronic control in conjunction with an accumulator injection systems provides a number of significant advantages over traditional fuel supply systems:

- ensuring flexible regulation of the law of supply and quantity of injected fuel in accordance with the specified load-speed operating conditions of the diesel engine;
- ensuring the necessary unevenness of fuel supply to the cylinders, including taking into account the individual characteristics of individual elements of the fuel supply system, their technical condition and the condition of the elements of the engine itself;
- shutdown of cylinders at partial loads;
- combining the control function with the engine diagnostic function through a system of control sensors, providing emergency engine protection.

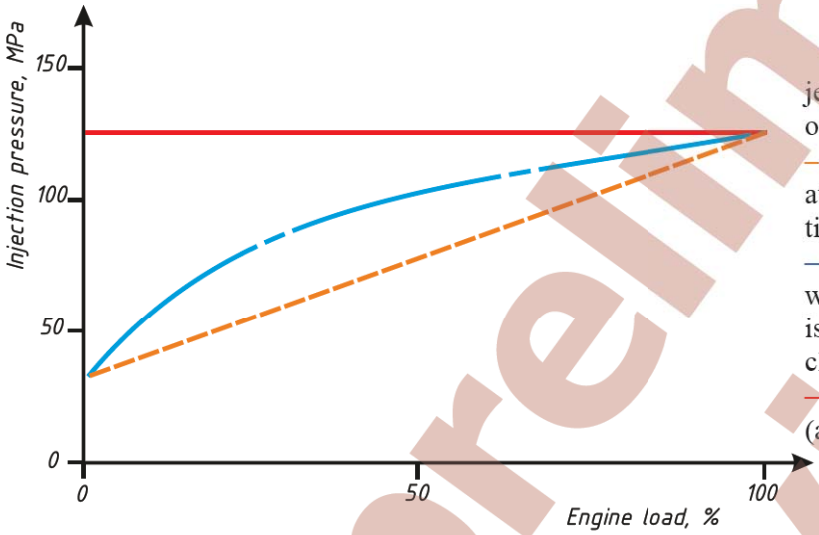


Figure 5.65 – Dependence of fuel injection pressure on engine load for various fuel supply systems:
– – – – direct-acting system when operating according to the screw characteristic (depending on the rotation speed);
—— – —— – direct action system when working according to the load characteristic (depending on the value of the cyclic supply);
——— – accumulator injection system (adapted from [9])

Of the variety of circuit solutions for accumulator injection systems on marine diesel engines, the most widely used are systems with pressure accumulators, made in the form of separate elements of large and small capacity.

5.7.1 Accumulator injection systems for low-speed diesel engines of the RT-flex series from Wärtsilä

Currently, the accumulator fuel injection system is used on low-speed diesel engines of the RT-flex series, produced by Wärtsilä, which inherited the developments of Sulzer, which released the first engine of this series in 2001. The system was designed to operate on heavy residual fuel HFO in accordance with ISO specification DIN 8217 (viscosity up to 700 cSt at 50°C) at temperatures up to 150°C.

The fuel system of RT-flex diesel engines is one of the components of the integrated engine management system, which also includes subsystems for controlling gas distribution, cylinder lubrication and starting air. The general diagram of the diesel control system of the RT-flex series is shown in Fig. 5.66.

The control functions of all subsystems are assigned to the microprocessor control unit WECS-9500 (Wärtsilä Engine Control System) (and its modifications), which is installed on each engine cylinder. All blocks are interconnected by a single system bus; if one block fails, its functions are automatically distributed among others. Thus, the failure of one block does not lead to failure of the cylinder.

For preliminary review

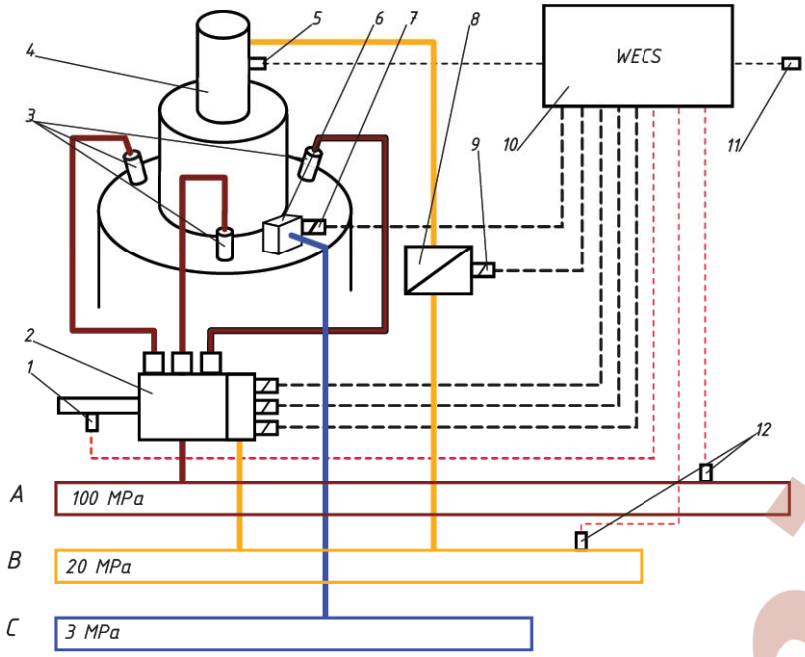


Figure 5.66 – Diagram of the control system for RT-flex series engines: 1 – injected fuel quantity sensor; 2 – fuel injection control unit; 3 – fuel injectors; 4 – exhaust valve drive mechanism; 5 – exhaust valve position sensor; 6 – start valve; 7 – electrohydraulic valve for controlling the starting valve; 8 – exhaust valve actuator; 9 – electrohydraulic valve for controlling the exhaust valve; 10 – electronic control unit; 11 – crankshaft position sensor; 12 – pressure sensors; *A* – fuel line; *B* – control oil line; *C* – air line (adapted from [49])

The system has two pressure accumulators: for control oil, which is pumped into the accumulator by axial plunger pumps at a pressure of 20 MPa, and for fuel, supplied to the accumulator under a pressure of up to 100 MPa by plunger pumps.

The general diagram of the engine fuel system is shown in Fig. 5.67.

Fuel under a pressure of 100 MPa flows through a non-return valve into a large-volume accumulator, which is a thick-walled pipe stretched along the entire engine. The fuel supply is controlled by an injection control unit installed on each cylinder, which receives control signals from the WECS 9500 microprocessor unit. Under the influence of these signals, high-speed control valves open or close fuel access to the standard fuel injectors, which inject fuel into the engine combustion chamber.

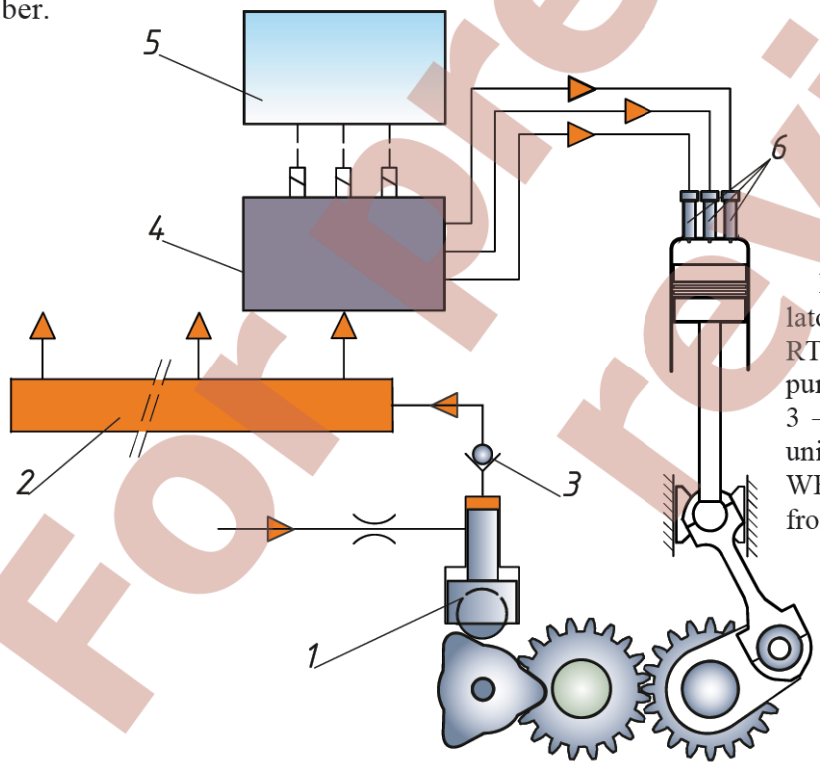


Figure 5.67 – Diagram of the accumulator injection system for engines of the RT-flex series from Wärtsilä: 1 – injection pump; 2 – pressure accumulator; 3 – check valve; 4 – fuel supply control unit; 5 – microprocessor controller WECS-9500; 6 – fuel injectors (adapted from [50])

Fuel and oil are pumped into the corresponding batteries by a pump unit, which is driven by the engine crankshaft through a system of intermediate gears (Fig. 5.68).

The design of the pump unit and its drive largely depends on the type of engine and its size. The general view and design of pumping units of RT-flex series engines is shown in Fig. 5.69.

For preliminary review

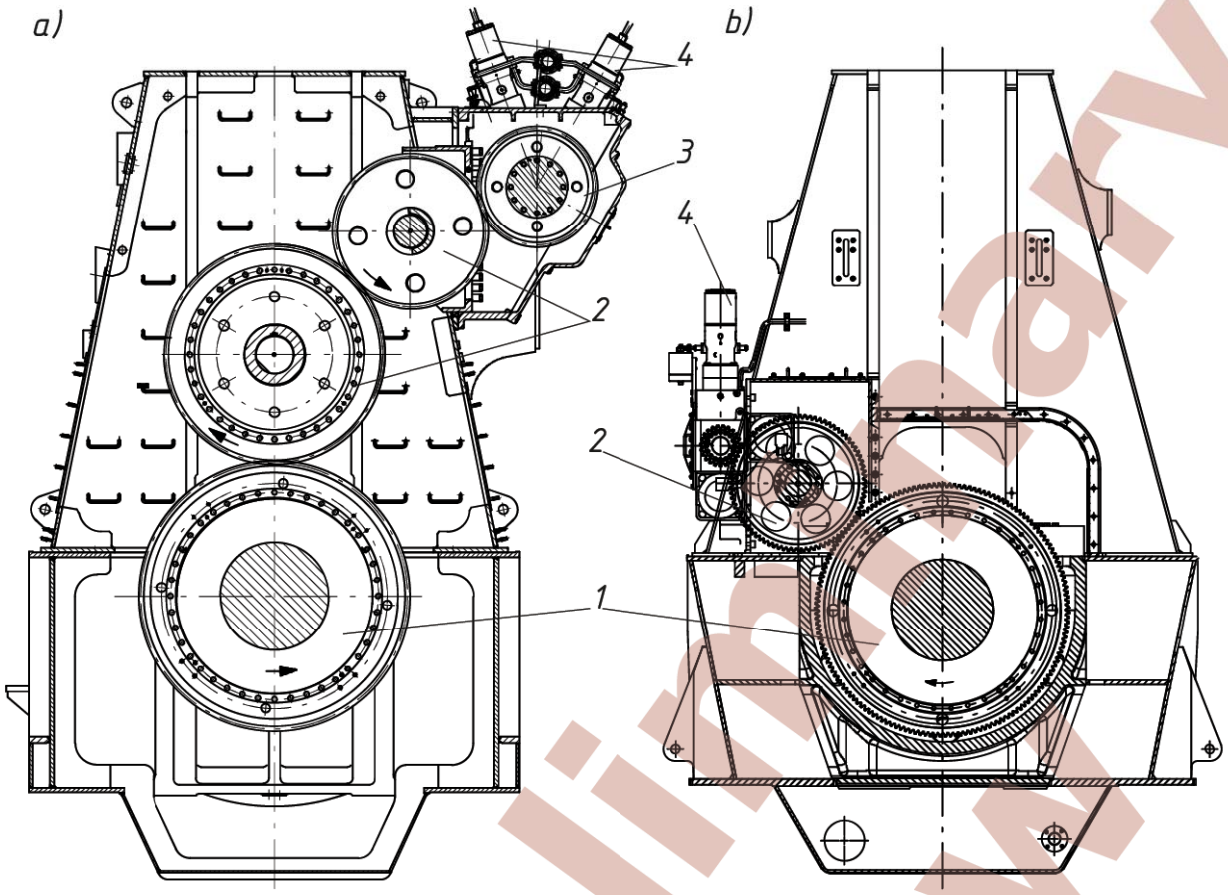


Figure 5.68– Drive mechanism of the RT-flex 96C (*a*) and RT-flex 50 D (*b*) diesel pump unit: 1 – drive gear; 2 – intermediate gears; 3 – drive gear; 4 – high pressure sections (adapted from [51, 52])

The pump drive unit has a housing in which the drive mechanism is mounted.

To supply oil to the control line, axial plunger pumps manufactured by Bosch or Dynex are used.

The pumps are driven by an intermediate gear through a splined shaft with a protective friction clutch. If one or more pumps jam, the friction clutch is destroyed, and the remaining pumps continue to operate.

The flow of the pumps through the servo drive system is controlled by the engine control unit.

On large diesel engines, the performance of the pumps is selected in such a way that, if one of them fails, the engine remains able to operate at full power. If more than one pump fails, the engine power must be reduced.

In early developments, pump blocks for large diesel engines were made double-row, with a V-shaped arrangement of high-pressure sections to reduce their size (Fig. 5.69 *a*, *b*). In later designs, preference is given to blocks with in-line sections, which are easier to maintain (Fig. 5.69 *c*, *d*). On large engines with a large number of cylinders, two pump drive units can be installed – at the bow and stern ends of the engine.

To drive the pumping sections, eccentric shafts are used (Fig. 5.69 *f*), and if it is necessary to increase the productivity of the sections, three-jaw drive washers are used, which are shifted relative to each other at an angle of 60° (Fig. 5.69 *e*).

In addition to pumps driven from the engine, the control oil system contains electrically driven pumps that supply the system with oil during diesel start-up, as well as in emergency situations.

High pressure sections used to supply fuel to the accumulator are plunger pumps of a traditional design (Fig. 5.70 *a*).

To control the performance of the sections, depending on the operating mode of the engine, spool-throttle regulation is used. For this purpose, a rack and pinion mechanism is provided for turning the plunger (Fig. 5.70 *b*), and on the plunger itself, throttling grooves with a variable cross-section are cut on both sides (Fig. 5.70 *c*, *d*). Depending on the angle of rotation of the plunger, the

For preliminary review

grooves block the filling channels in the sleeve partially or completely, changing the flow area of the filling hole. This increases or decreases the fuel throttling resistance at the entrance to the space above the plunger.

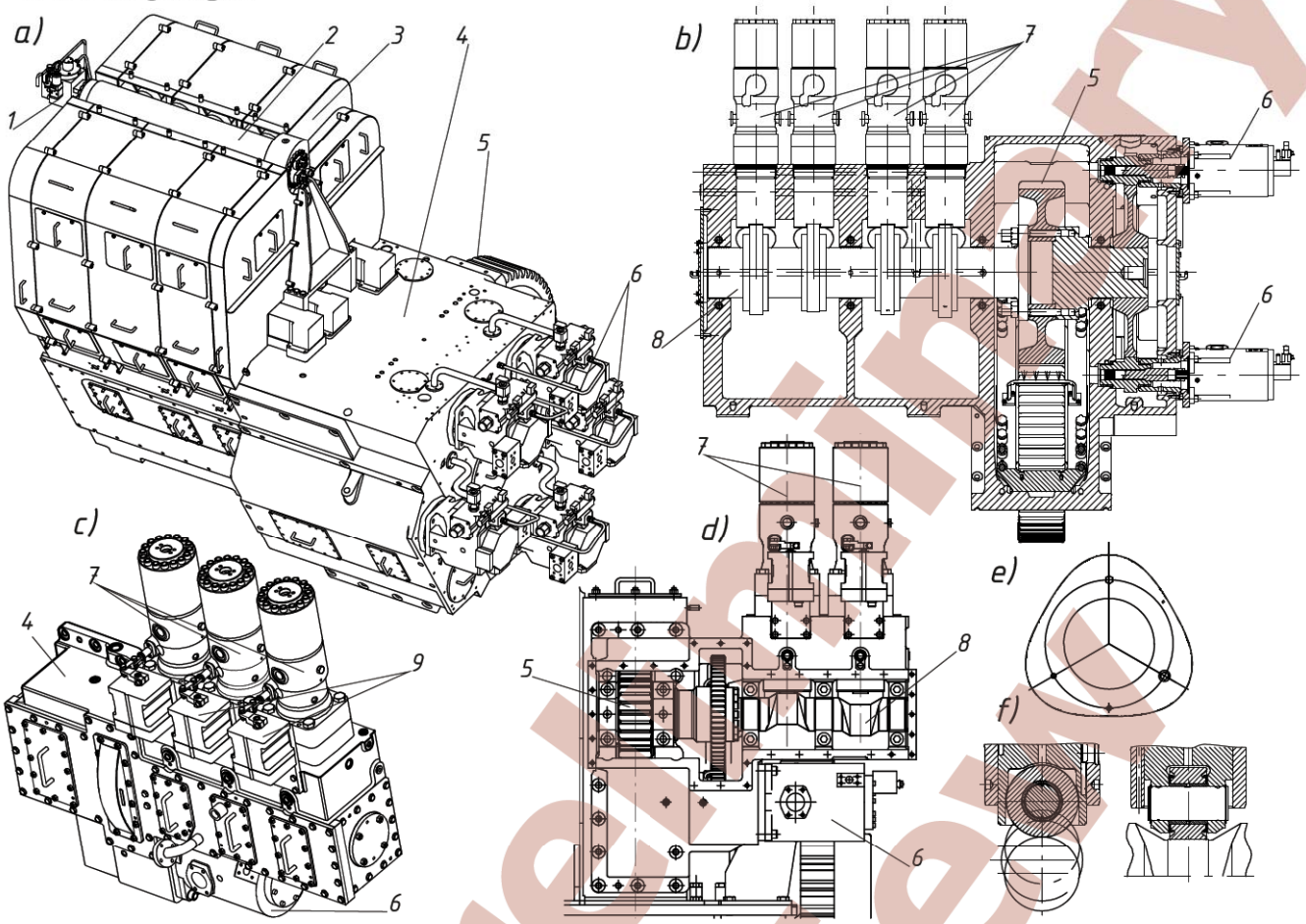


Figure 5.69 – Pump blocks for diesel engines RT-flex 96C and RT-flex 50 D: 1 – pressure limiter; 2 – intermediate accumulator; 3 – block of high pressure sections; 4 – housing with drive mechanism; 5 – drive gear; 6 – axial plunger pumps of the control oil system; 7 – fuel injection pump sections; 8 – cam shaft of the drive of the high pressure sections; 9 – servomechanism for controlling the supply of high-pressure sections; *a* – general view of the pump unit RT-flex 96C diesels; *b* – drive mechanism for the pump unit RT-flex 96C diesels; *c* – general view of the pump unit RT-flex 50T diesels; *d* – drive mechanism for the pump unit RT-flex50T diesels; *e* – fuel injection pump drive cam with three protrusions of the RT-flex 96C diesel; *f* – eccentric drive of the diesel fuel injection pump RT-flex 50T engine (adapted from [51, 52])

The performance of the pumping sections is changed by moving the racks using a servo drive mechanism that receives signals from the engine control unit.

The use of throttle control has made it possible to reduce the cost of driving the pump sections, since only the volume of fuel that, in accordance with the engine operating mode, must be supplied to the accumulator to compensate for the volume spent on injection, enters and is compressed above the plunger space.

If the pump section fails, its pusher can be locked in the upper position using a special locking mechanism. The failure of one section does not lead to the need to reduce engine power.

The total number of high-pressure sections installed on the block depends on the number of engine cylinders and its declared power.

The intermediate pressure accumulator is a thick-walled pipe, closed at the ends with lids.

Connecting fittings with check valves mounted in them are attached to the accumulator housing, through which fuel from the pump sections flows through high-pressure pipes (Fig. 5.71 *a*).

In addition to the high-pressure cavity, the housing of the accumulator has a number of drainage channels through which fuel leaks are diverted to the leakage collection tank. In addition to the

For preliminary review

connecting fittings, safety valves adjusted to a pressure of 125 MPa and fittings for connecting measuring instruments are installed on the accumulator housing (Fig. 5.71 *b*).

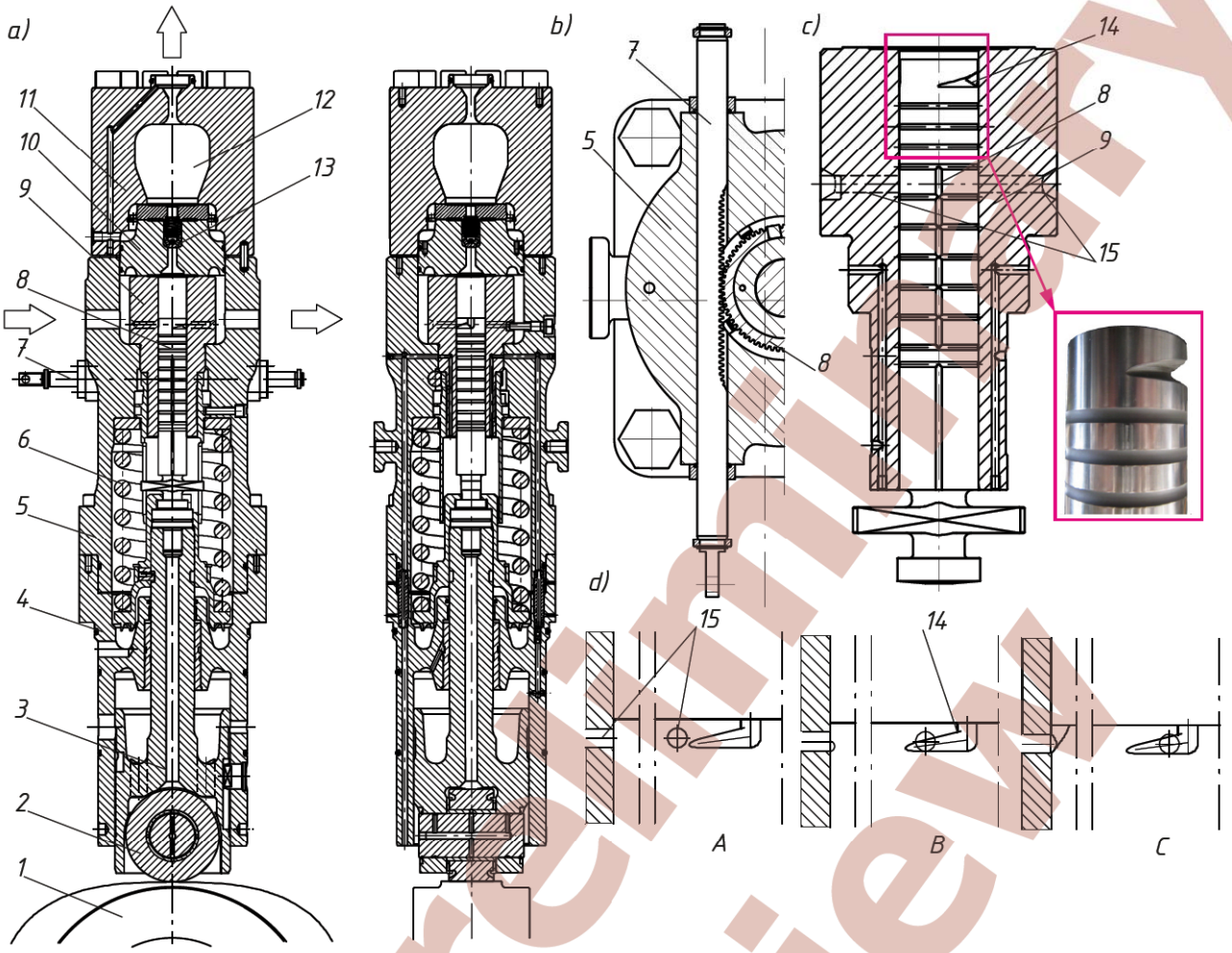


Figure 5.70 – High-pressure section of the RT-flex 96C diesel engine (*a*), rack-and-pinion plunger rotation mechanism (*b*), plunger pair (*c*) and uel pump section delivery control scheme (*d*). 1 – drive cam; 2 – pusher roller; 3 – pusher; 4 – pusher housing; 5 – high pressure section housing; 6 – return spring; 7 – rack; 8 – plunger; 9 – plunger bushing; 10 – check valve housing; 11 – cover; 12 – buffer cavity; 13 – check valve; 14 – throttling groove; 15 – filling hole. *A* – zero supply; *B* – partial supply; *C* – full supply (adapted from [51, 52])

At the ends of the accumulator housing there are covers, to one of which the pressure limiter regulator is attached, and to the other – the connecting fittings of the high-pressure pipelines for supplying fuel to the main accumulator.

In order to ensure greater reliability, two parallel high-pressure lines are laid between the intermediate and main accumulators with shut-off valves installed on them, which allows them to be put into operation both together and separately. The capacity of each line allows the engine to operate at full power.

Between the housing and the covers, special lip seals made of polymer materials are installed, which are pressed against the end surfaces of the cover and housing by the force of fuel pressure.

On engines with a small cylinder diameter, an intermediate accumulator may not be installed, and the fuel goes directly from the high-pressure sections to the main accumulator.

The pressure limiter regulator serves to maintain a constant high pressure in the fuel accumulation system.

The pressure regulation limit is 50...105 MPa, but the recommended operating range is 60...80 MPa.

For preliminary review

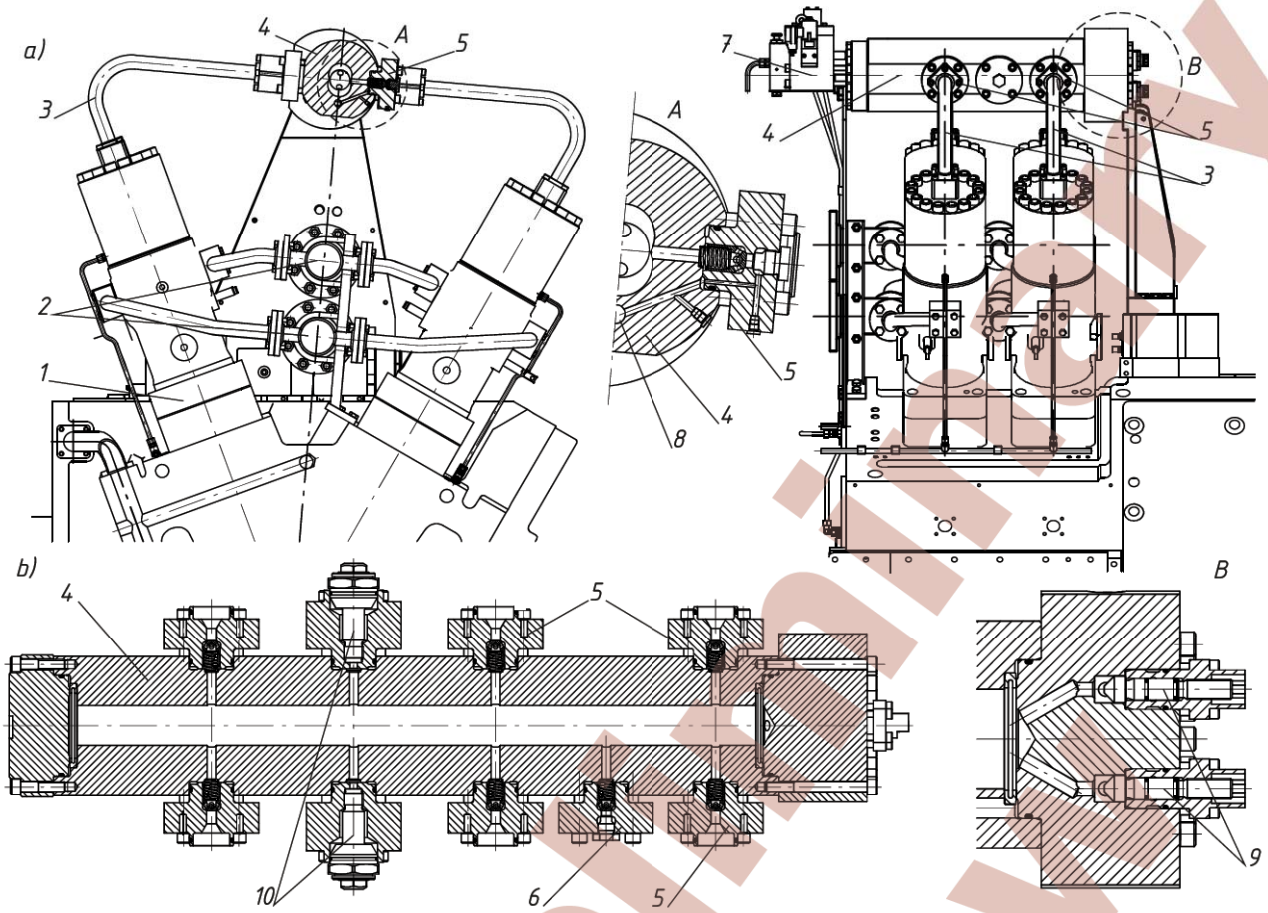


Figure 5.71 – Intermediate pressure accumulator for RT-flex 96C diesel engines: *a* – placement of the accumulator on the pump block; *b* – longitudinal section of the accumulator: 1 – high pressure pump section; 2 – fuel supply lines to the high pressure sections; 3 – high pressure pipelines; 4 – intermediate accumulator housing; 5 – connecting fitting with a check valve; 6 – fitting for connecting measuring instruments; 7 – pressure limiter regulator; 8 – drainage channel; 9 – connecting fittings for high-pressure pipelines for supplying fuel to the main accumulator with disconnect valves; 10 – safety valves (adapted from [51, 52])

The main working element of the regulator is the needle valve, which separates the high-pressure cavity and the fuel drain line (Fig. 5.72).

In the closed position, the valve is held by the force created by a hydraulic piston, the loading oil of which is supplied through a throttling device from the control line. The required amount of force is achieved by selecting the diameter of the hydraulic piston.

The above-piston cavity is separated from the oil drain cavity by a spool valve, which is loaded on one side by a spring and on the other by a piston communicating with the high-pressure fuel cavity. If the pressure in the fuel cavity increases above the set value, the piston, overcoming the force of the spring, moves the spool valve upward, thereby connecting the above-piston cavity with the oil drain cavity. If the oil flow through the spool is greater than the oil flow through the throttling device, the pressure in the above-piston cavity will decrease, the needle valve will open and release part of the fuel into the drain line. As a result, the pressure in the system will decrease, the spool will return to its original position, and the needle valve will close.

The required pressure in the fuel accumulation system is set by pre-tightening the spool loading spring. To do this, place a corresponding set of spacer washers under the pre-tightening screw.

An emergency pressure relief valve in the accumulator is located on the pump housing, which connects the cavity above the hydraulic piston to the oil drain line. The valve has an electromagnetic and manual drive.

On small engines that do not have an intermediate accumulator, the limiter valve is installed directly on the main accumulator (Fig. 5.73 *b*).

For preliminary review

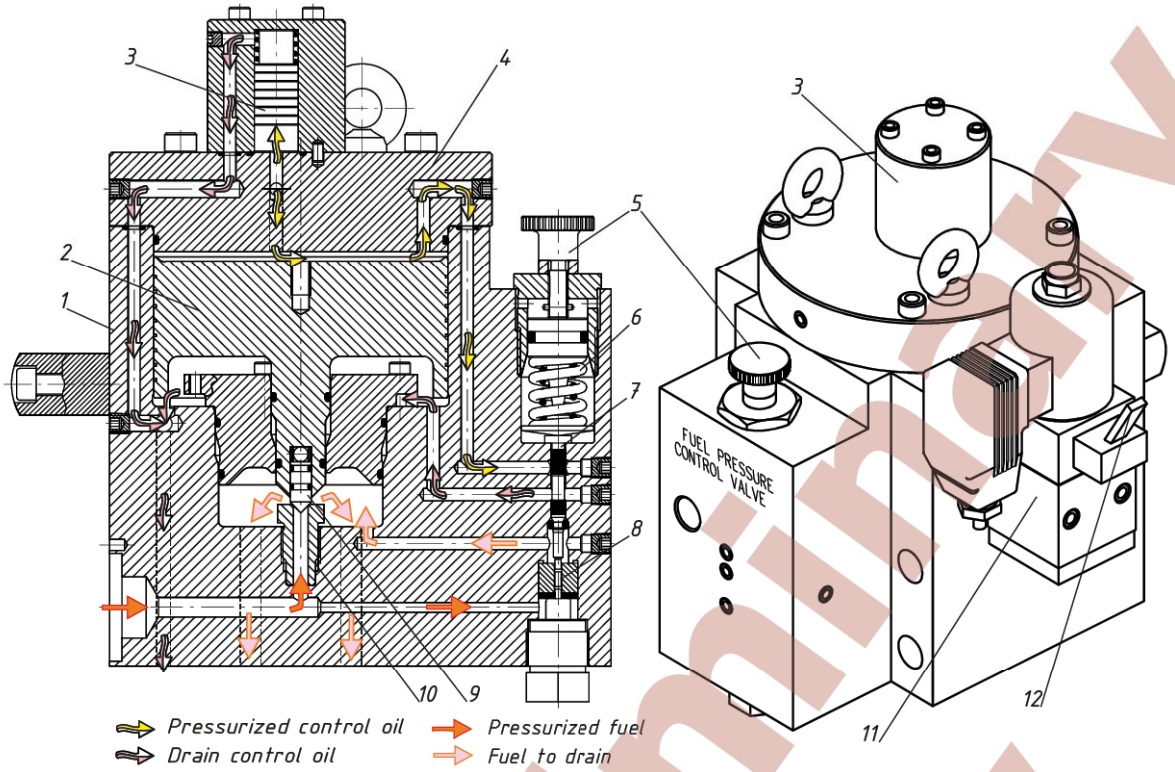


Figure 5.72 – Pressure limiter regulator for diesel engines of the RT-flex series: 1 – regulator housing; 2 – hydraulic piston; 3 – oil pressure accumulator; 4 – cover; 5 – screw for installing pre-tensioning of the loading spring; 6 – loading spring; 7 – spool; 8 – piston; 9 – needle valve; 10 – needle valve seat; 11 – solenoid valve for emergency pressure relief; 12 – pressure release lever (adapted from [51, 52])

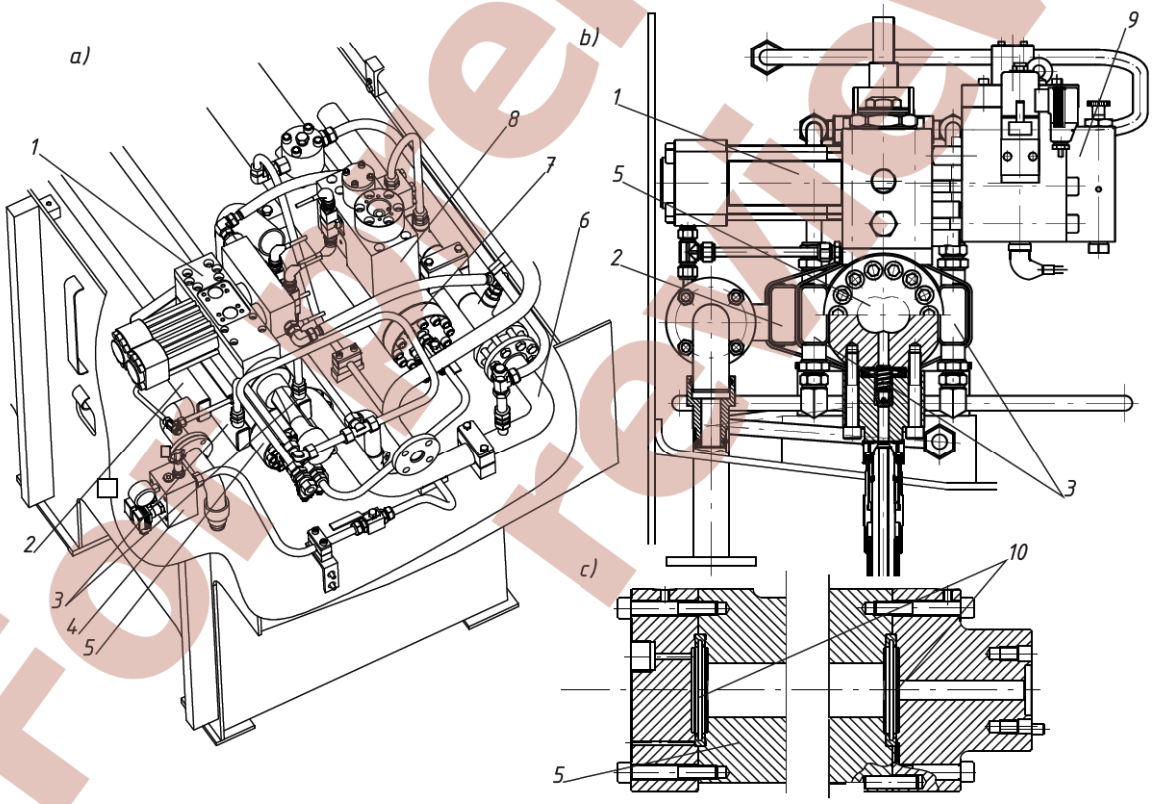


Figure 5.73 – Main module for controlling the fuel supply and gas distribution of the RT-flex 50 D diesel engine (*a*), main fuel accumulator with a heating system and fuel supply control unit (*b*), cross-section of the main accumulator (*c*): 1 – fuel supply control unit; 2 – low pressure fuel heating line; 3 – main accumulator heating lines; 4 – control oil pressure regulator; 5 – main accumulator; 6 – waste oil collection line; 7 – oil accumulator of the exhaust valve hydraulic drive system; 8 – exhaust valve actuator; 9 – pressure limiter regulator; 10 – seals (adapted from [53])

For preliminary review

The fuel supply and gas distribution control platform is a metal box located on the upper platform of the engine in close proximity to the cylinder covers. Inside the box there is a main fuel accumulator with fuel supply control units installed on it, an accumulator of the hydraulic drive system with exhaust valve actuators installed on it, and a control oil line (Fig. 5.73 *a*). In addition, a drain line is laid here to collect used oil in the hydraulic drive system and control oil, as well as a line for collecting fuel leaks.

The main accumulator is similar in design to the intermediate accumulator, but its length is almost equal to the length of the engine (Fig. 5.73 *b*, *c*).

The accumulator housing has milled areas for installing fuel supply control units.

To supply fuel to the accumulator, fittings with check valves are installed in the side covers, to which high-pressure fuel lines are connected.

To maintain the required fuel temperature, heating lines tightly adjacent to it, made in the form of rectangular pipes, are installed along the accumulator. Steam at the appropriate temperature is passed through the pipes.

If the length of the engine is sufficiently long, the main accumulator is made as a composite accumulator. The individual sections are connected to each other using intermediate inserts (Fig. 5.74).

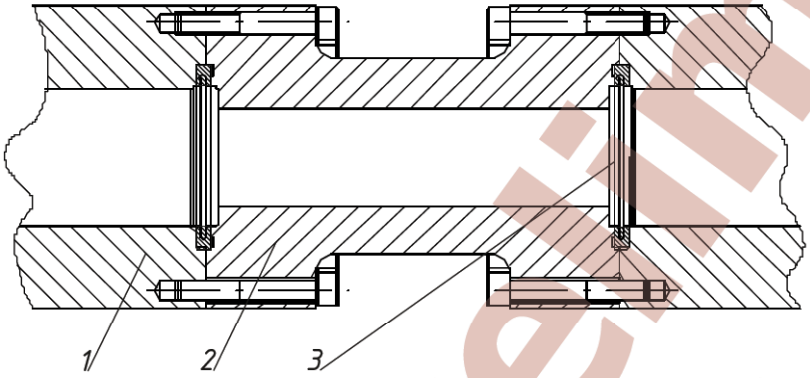


Figure 5.74 – Connection of main accumulator sections: 1 – accumulator housing; 2 – intermediate insert; 3 – seals (adapted from [51])

On engines that have nine cylinders and above, two batteries are usually installed, which are connected to each other by special pipelines, each of which has shut-off valves and is capable of ensuring full passage of fuel between the batteries.

The large volume of the main accumulator allows for fairly stable pressure throughout the entire injection, even without the use of gas or spring pressure compensators. Individual injections do not cause significant pressure fluctuations in the accumulator, which can significantly affect the fuel supply process (Fig. 5.75).

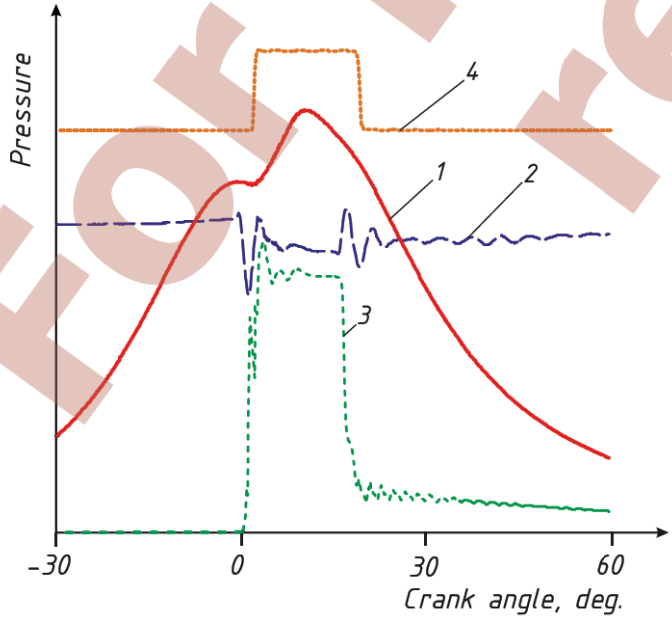


Figure 5.75 – Fuel injection process with the accumulator fuel supply system of the RT-flex series diesel engine. 1 – pressure curve in the cylinder; 2 – pressure in the main accumulator; 3 – pressure in front of the nozzle; 4 – fuel injector needle lift (adapted from [54])

For preliminary review

The injection control unit (ICU) is the most critical element of the fuel system of RT-flex series engines.

Depending on the size of the working cylinder, engines of this series are equipped with two or three fuel injectors per cylinder. Accordingly, the control unit consists of two or three control modules.

Each module includes two high-speed valves – a paired valve for controlling the fuel supply to the injector with a hydraulic drive, and a spool valve with an electromagnetic drive that controls this drive.

The hydraulic diagram of the injection control module is shown in Fig. 5.76.

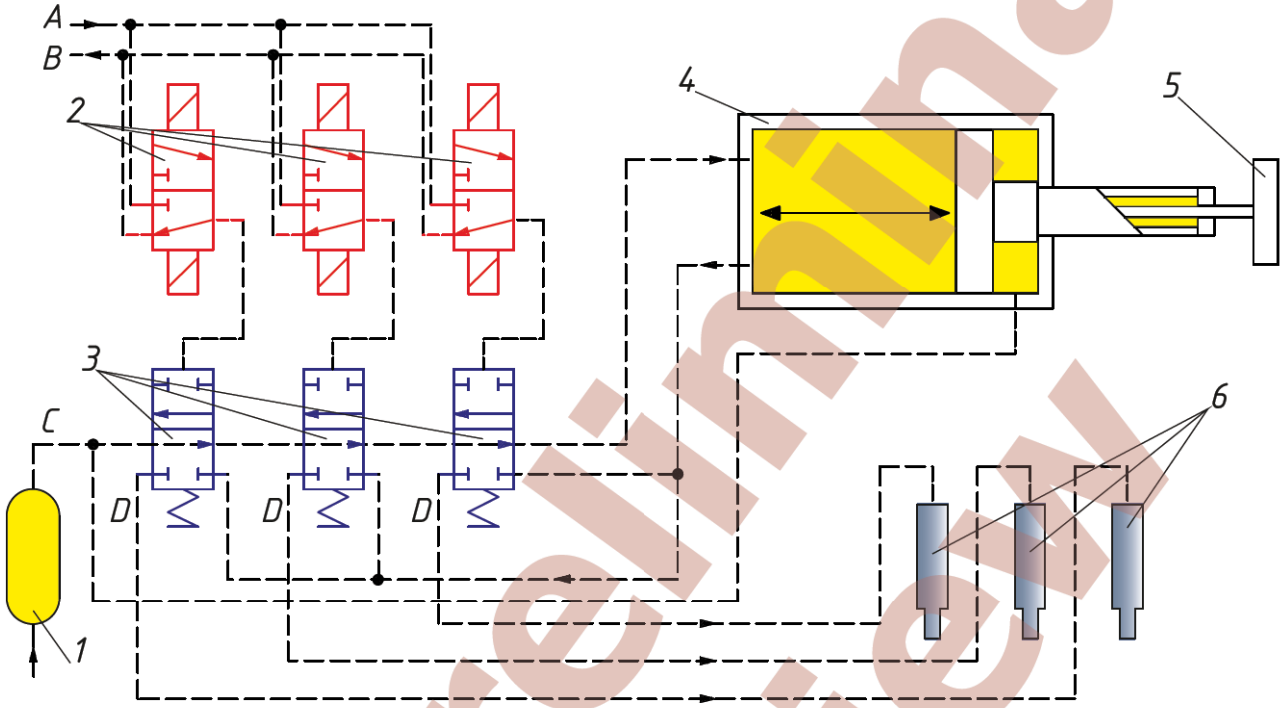


Figure 5.76 – Hydraulic diagram of the injection control module: 1 – accumulator; 2 – spool valve with electromagnetic drive; 3 – valve for controlling the fuel supply to the injector; 4 – measuring cylinder; 5 – fuel consumption sensor; 6 – fuel injectors. *A* – control line; *B* – drain line; *C* – high pressure fuel supply; *D* – fuel to injectors (adapted from [55])

The electric valve is controlled by the WECS-9500 microprocessor module.

For the correct selection of injection parameters, there is feedback between the control unit and the microprocessor module, which is implemented through a sensor for monitoring the amount of injected fuel.

The principle of determining fuel consumption is based on recording the movement of a special measuring piston located in the control unit housing.

The operating principle of the injection control module is shown in Fig. 5.77.

In the absence of a signal from the electronic control unit, fuel, under high pressure, flows from the accumulator into the cavity of the measuring cylinder. To the cavity, on the side of the measuring rod, fuel flows directly from the accumulator, and into the cavity free from the rod – through the open spool valve.

Thus, the same pressure acts on the piston on both sides, however, given that the area of the piston on the side of the measuring rod is smaller by the cross-section of the rod itself, a force of greater magnitude acts on the free side of the piston. Under the influence of the force difference, the piston moves in the direction of the rod, moving the sensor screen to its extreme position.

In addition, from the cavity of the measuring cylinder, fuel enters the cavity above the main supply control valve, but since the valve is closed at this moment, fuel is not injected into the cylinder.

For preliminary review

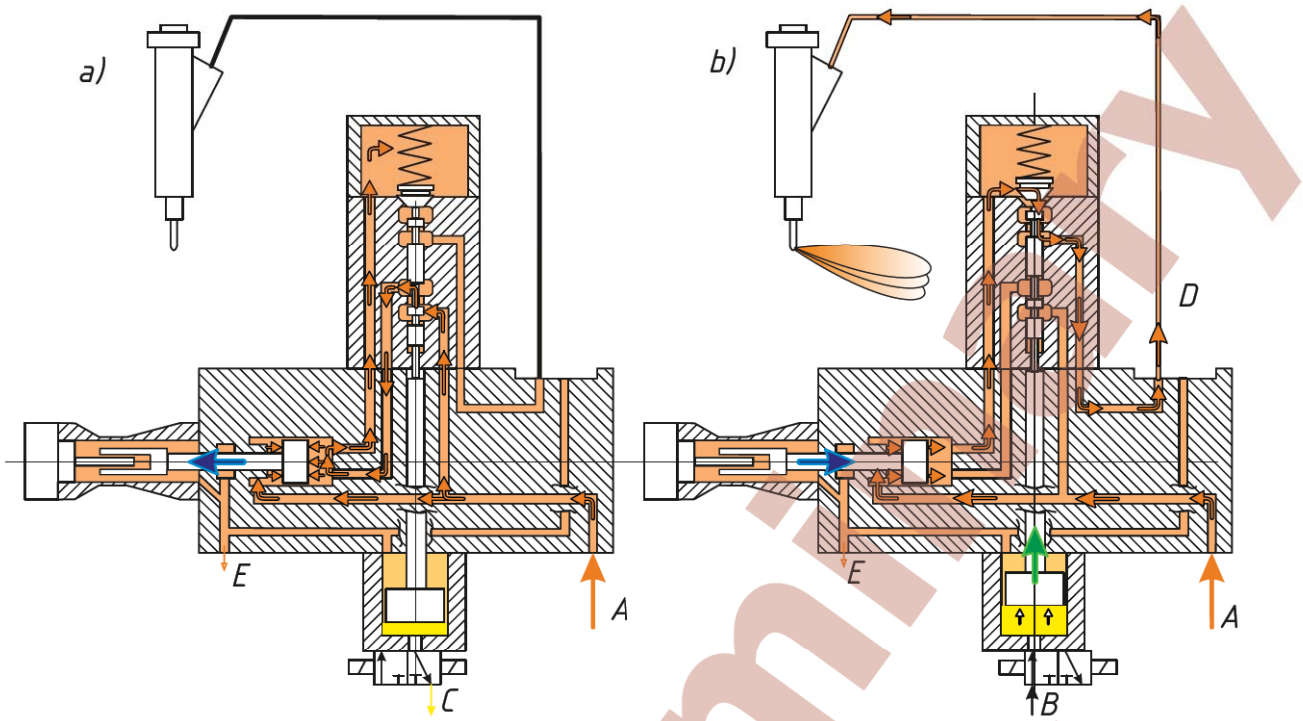


Figure 5.77 – Fuel supply control module for diesel engines of the RT-flex series: *a* – filling the measuring cylinder; *b* – fuel injection. *A* – fuel supply under pressure from the accumulator; *B* – oil supply to the hydraulic drive of the fuel control valve; *C* – draining oil from the hydraulic drive system of the fuel control valve; *D* – fuel supply to the injector; *E* – draining fuel leaks (adapted from [9])

When an electrical signal is received, the spool valve moves and supplies oil from the control line to the piston cavity of the hydraulic drive of the supply control valve. Oil pressure causes the actuator piston to move the valve stem upward. As a result, the lower valve blocks the access of fuel to the free cavity of the measuring piston, and the upper valve opens fuel access to the fuel injector. With this valve position, fuel enters the measuring cylinder only from the rod side, causing the piston to move, displacing fuel from the free cavity of the measuring piston through the open upper valve to the fuel injector. When the pressure reaches 37.5 MPa, the fuel injector opens and fuel injection into the engine combustion chamber begins.

The movement of the measuring piston is converted by the fuel flow sensor into an electrical signal and transmitted to the control unit, which calculates the amount of fuel supplied to the cylinder. Based on this, a decision is made to continue or terminate the supply.

When an electric signal to interrupt the injection is applied to the spool valve, the spool moves, connecting the cylinder cavity with the drain line. The oil from the main valve hydraulic drive cavity is drained, and it returns to its original position – fuel injection stops. Then the whole cycle repeats.

The fuel supply control unit is attached directly to the fuel accumulator housing (Fig. 5.73 *b*).

The general view and design of the injection control unit for RT-flex series engines is shown in Fig. 5.78.

Given the high operating temperature of the accumulator and control unit, to protect the fuel flow sensor from high temperatures, it is placed in a separate housing located outside the unit. In some engines, to improve the operating conditions of the sensor, cooling fins are applied to its housing.

The housings of the supply control valves are made in the form of separate elements, which are mounted on the common housing of the block in accordance with the number of injectors per cylinder.

A measuring cylinder is placed in the block housing, with the help of which the total amount of fuel supplied to the injectors by all modules included in the control unit is determined. In addition, there is a system of channels for supplying fuel to individual modules, as well as a system for col-

For preliminary review

lecting and draining leaks common to all elements of the block.

On the opposite side, hydraulic drive cylinders for the supply control valves and spool valves with an electromagnetic drive are attached to the housing.

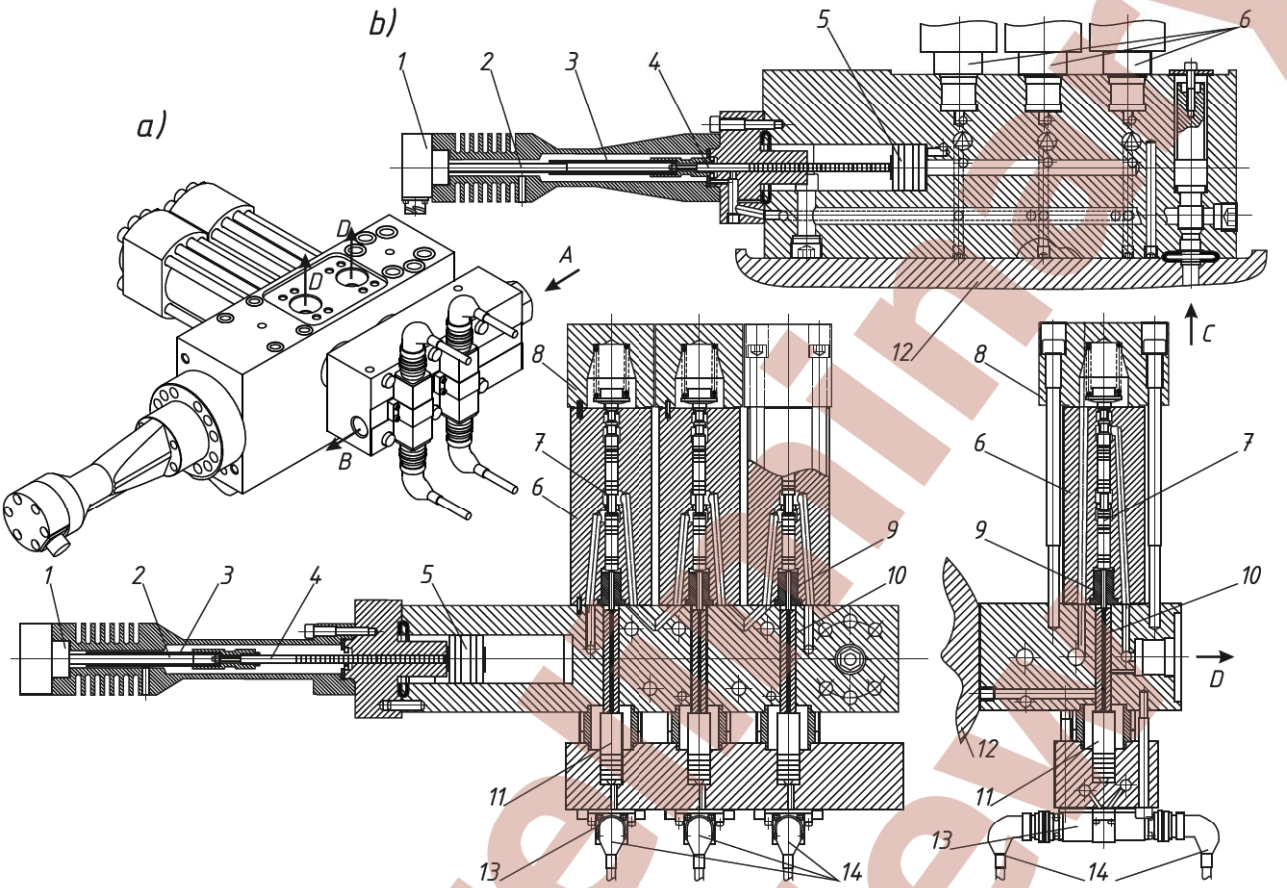


Figure 5.78 – Figure 5.78 – General view of the fuel supply control unit RT-flex 50T diesels (*a*) and the design of the fuel supply control unit RT-flex 96C diesels (*b*): 1 – fuel consumption sensor; 2 – sensitive element of the fuel consumption sensor; 3 – movable screen; 4 – measuring piston rod; 5 – measuring piston; 6 – supply control valve housing; 7 – supply control valve; 8 – supply control module cover; 9 – guide sleeve; 10 – intermediate rod; 11 – hydraulic piston of the supply control valve drive; 12 – accumulator housing; 13 – spool valve with electromagnetic drive; 14 – connecting connectors. *A* – control oil supply; *B* – control oil drain; *C* – fuel supply from the accumulator; *D* – fuel supply to the injectors (adapted from [55])

High-speed spool valves with an electromagnetic drive are made according to the so-called bistable circuit. This means that when an electrical signal is sent to the valve, the regulating element, in this case the spool, moves to a certain position and remains in it even after the signal is removed until another signal is sent to the actuator, which moves the regulating element in the opposite direction position. Thus, the regulatory has two stable positions.

The general design of the valve is shown in Fig. 5.79.

The central housing houses a spool and a system of channels for supplying and discharging control oil. The spool moves under the action of electromagnets located on opposite sides of the housing. The control current is supplied to the electromagnet coils through the connecting connectors.

The valve response speed does not exceed 1…2 ms. This is achieved, on the one hand, by a very small stroke of the spool, which does not exceed 0.3 mm, and, on the other hand, by supplying the coil with a large current, which is 50...60 A. To protect the coil from excessive overheating, the current is supplied to it in the form of short pulses with a duration of 1...3 ms. The maximum permissible pulse time does not exceed 4.5 ms.

When preparing the engine for starting, in order to move all bistable valves to the «no supply»

For preview only

position, the electronic control unit sends corresponding pulses to them every 10 s.

Together with the control valve, the spool valve forms a separate fuel control module. The modules included in the fuel supply control unit can provide both synchronous fuel supply to all injectors simultaneously, and an individual injection mode for each injector. For example, in engines with three injectors per cylinder, to ensure minimum stable speeds, the control system provides the ability to sequentially turn off each of the three injectors within one cylinder or operate on two or even one injector (Fig. 5.80).

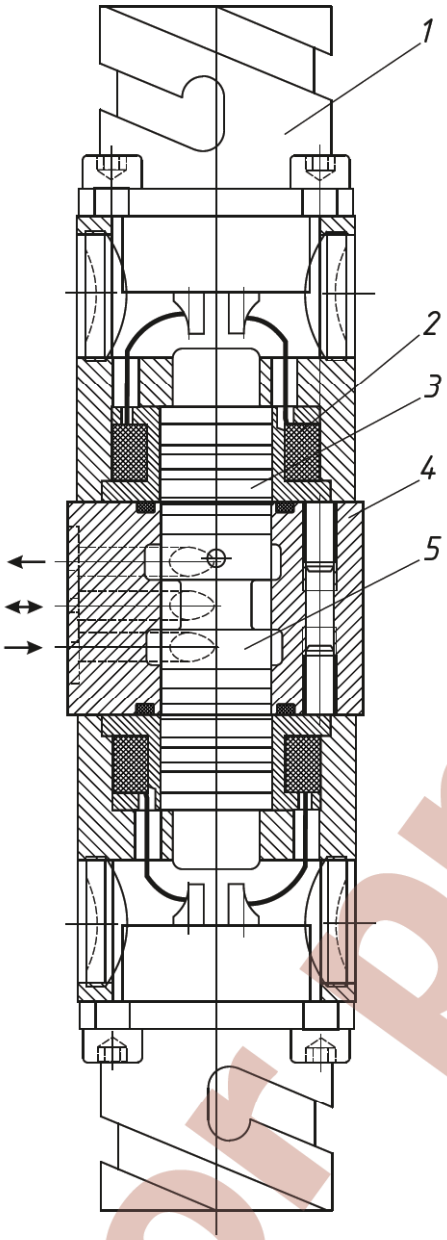


Figure 5.79 – Quick-acting spool valve with an electromagnetic drive: 1 – electrical connector; 2 – solenoid; 3 – magnetic core; 4 – valve housing; 5 – spool

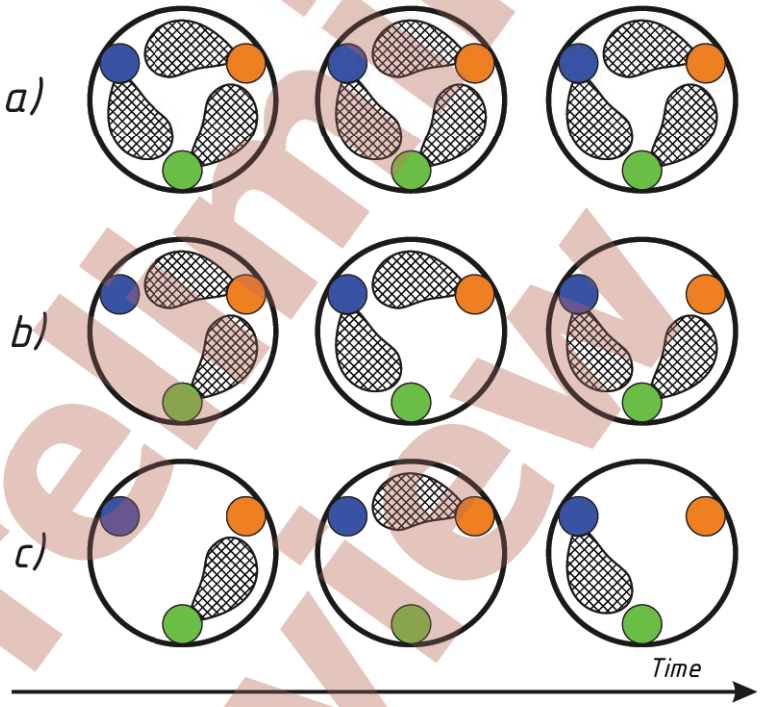


Figure 5.80 – Fuel injection modes provided by the accumulator supply system of RT-flex 96C series engines: *a* – full load, all injectors are working; *b* – average load, only two fuel injectors work; *c* – low load, one injector is working (adapted from [54, 56])

Disabling one or two injectors increases the duration and stability of the flow of the remaining injectors. This ensures stable operation of the engine at a speed of 10...12% of the nominal speed.

To avoid overheating and coking of the fuel injectors, the fuel injectors are switched on one by one, every 20 minutes. Between individual cylinders there is a delay in switching injectors of 10 seconds, which makes it possible to reduce the smoke of exhaust gases due to the injection of a portion of cooler fuel into the cylinder by a new injector.

5.7.2 Accumulator injection systems of low-speed diesel engines with direct fuel supply control

Almost immediately after the launch of the first generation RT-flex series engines, Wärtsilä began work on the creation of a second generation of Common Rail systems using direct injection in-

For preliminary review

jectors. This made it possible to increase the dosing accuracy, especially at low and medium loads, while simultaneously reducing the number of elements included in the system. In this case, the element base was partially used, developed for first-generation systems, and partially for four-stroke medium-speed engines. The system was initially developed for low-speed engines with a cylinder diameter of up to 50 cm. The first 6RT-flex35 engine equipped with the new system was released in 2011. These developments were subsequently inherited by Winterthur Gas and Diesel (WinGD), which became the successor to Wärtsilä in the low-speed marine engine sector. Based on second-generation technology, WinGD launched the production of a new line of WX series engines with cylinder diameters from 35 to 72 cm.

MAN Diesel & Turbo SE developed its own version of Common Rail fuel systems for low-speed engines with a cylinder diameter of up to 50 cm. The first commercial engine, the 6S35ME-C9-CR, equipped with this system, was built by subsidiary MOL Mitsui O.S.K. Kinkai Ltd. in 2018.

Common rail engines from both manufacturers are designed to operate on liquid fuels in accordance with ISO 8217 DMA, DMZ and DMB, as well as on HFO up to 700 cSt (in accordance with ISO-F-RMK 700). It is also possible to create on their basis engines running on alternative fuels, which we discussed earlier. To a large extent, the systems are similar both in circuit and elemental solutions, for this reason it makes sense to consider them together, focusing only on the differences.

The fuel pressure accumulator in both systems is a single-wall pipe into which fuel is pumped by several high-pressure fuel pumps. The number of installed pumps depends on the number of working cylinders, but not less than two, to ensure that the engine can operate if one of the pumps fails. Small engines (35...50 cm) use multi-section fuel pumps manufactured by L'Orange with throttle performance regulation, which have proven themselves on medium-speed engines (will be discussed in the following chapters). They are distinguished by their reliability, small size and well-established production and service, which significantly reduces the cost of the fuel system. The pumps are designed to create pressures of 160 MPa, while for low-speed engines the operating pressure is limited to 120 MPa. To obtain the specified pressures, the pump rotation speed must be within the range of 2300...3000 min^{-1}. To obtain a given frequency, WinGD engines use a drive from the crankshaft with an overdrive (Fig. 5.81), while MAN uses an independent electric drive.

The electric drive of the pumps is carried out from the on-board network through a frequency regulator, which makes it possible to coordinate the operating modes of the pumps with the operating mode of the engine. The amount of fuel supplied by the pump is controlled by a control unit (throttle valve) installed on the suction side.

Fuel is supplied to the pumps through automatic backwash filters installed in series, designed for a cleaning fineness of 50 and 6 μm.

Lubricating oil is supplied to each pump by taking it from the engine's circulating lubrication system. If higher pressure is required to lubricate the pumps, additional oil pumps are installed.

In large-size engines (52-72 cm) from WinGD, pump sections designed for the first generation RT-flex systems, the design and operating principle of which we discussed earlier, are used to supply fuel to the accumulator.

Each high-pressure pump is connected to the fuel accumulator by separate pipes with double walls through a check valve installed in a special unified housing directly on the fuel accumulator (Fig. 5.82). This is a normally open valve, which is designed to block reverse flow from the accumulator in the event of a leak from the fuel supply pipe from the high pressure pump (Fig. 5.83). During normal operation, fuel pressure opens the valve, overcoming the force of the spring, allowing fuel to flow into the accumulator. This ensures that the fuel system can continue to operate if one of the pipes is damaged or one of the fuel pumps fails.

Combination pressure control valves are installed on the end caps of the accumulators (MAN installs two valves that duplicate each other) which act as mechanical safety valves and as controlled fuel pressure regulators in the accumulator. These same valves are used to remotely relieve pressure during an emergency engine shutdown and to circulate heated fuel through the system when the engine is stopped.

For preliminary review

The valve design is similar to those used in the first generation RT-flex engines. The opening pressure of the valves lies in the range of 115.0...120.0 MPa and is set by the force of the spring, which acts on the shut-off needle. If the system for regulating the performance of high-pressure fuel pumps fails, the valve needle opens slightly, ensuring throttling of the fuel drained from the accumulator. As a result, the pressure in the accumulator stabilizes at 70.0...90.0 MPa. In this condition, the engine can be operated at reduced load until the loss of pump performance control is corrected. Thus, the valve is used as a backup valve in case of failures in the pump control mechanism.

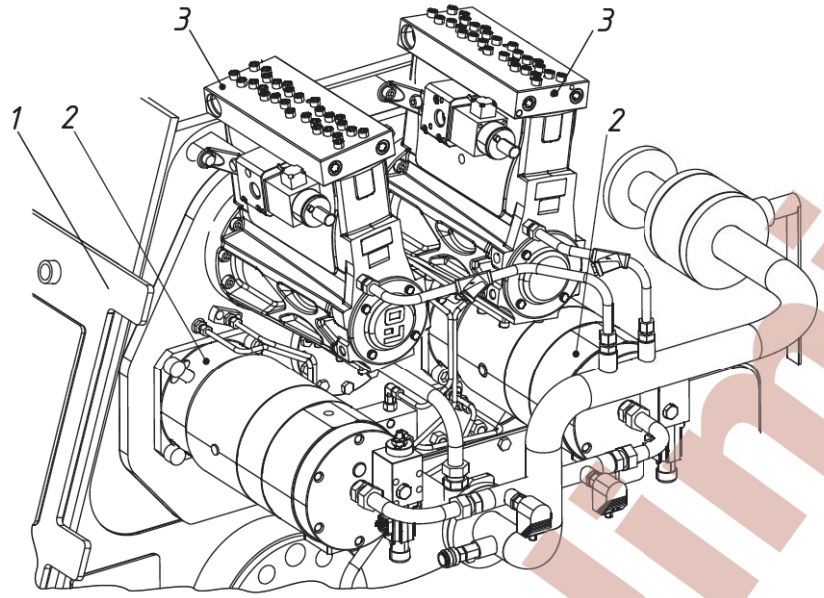


Figure 5.81 – Location of multi-section fuel and axial plunger oil pumps on the X-35 series engine from WinGD: 1 – engine crankcase; 2 – oil pump of the control system; 3 – high pressure fuel pumps from L'Orange (adapted from [57])

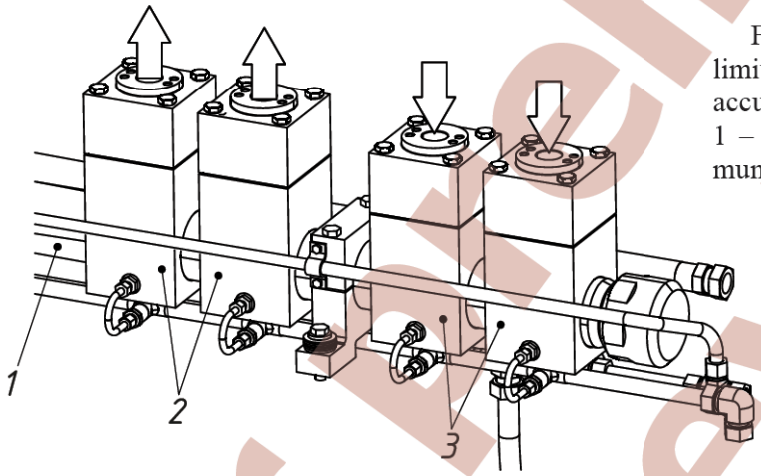


Figure 5.82 – Location of maximum flow limitation valves and check valves on the fuel accumulator of WinGD X-35 series engines: 1 – accumulator; 2 – valves for limiting maximum flow; 3 – check valves (adapted from [57])

Fuel is supplied to each injector through flow limiting valves (FLV) built into the connecting block mounted on the accumulator (Fig. 5.82). An example of such a valve in a housing unified with the check valve housing of X-35 series engines from WinGD is shown in Fig. 5.84. The main purpose of the limiter valve is to stop the fuel supply to the injector in the event of depressurization of the high pressure fuel line or the injector needle hangs open. Closing the valve shuts off the damaged high pressure line, allowing the engine to continue running.

The valve's shut-off cone is connected through a rod to a hydraulic piston, which fits tightly to a cylinder in the valve housing. The above-piston cavity communicates with the accumulator cavity through the throttle hole. In the absence of injection, fuel from the accumulator flows through this channel into the above-piston cavity, and the piston, under the action of the return spring, takes its lowest position (Fig. 5.81).

When injection begins, the pressure in the above-piston cavity decreases, since the throttle hole does not have time to compensate for the fuel flow through the fuel injector. Under the influence of the pressure difference, the piston begins to move, pushing fuel out of the space above the piston. The volume of the cavity above the piston corresponds to 130...170% of the maximum cyclic flow. If the injector has not stopped flowing by the time the piston reaches its extreme position, the shut-

For preliminary review

off cone of the limiter valve sits on the seat and cuts off the fuel supply to the damaged high-pressure line. If the fuel supply stops before the valve sits on the seat, the piston, under the action of a spring, returns to its original position during the period of time between injections.

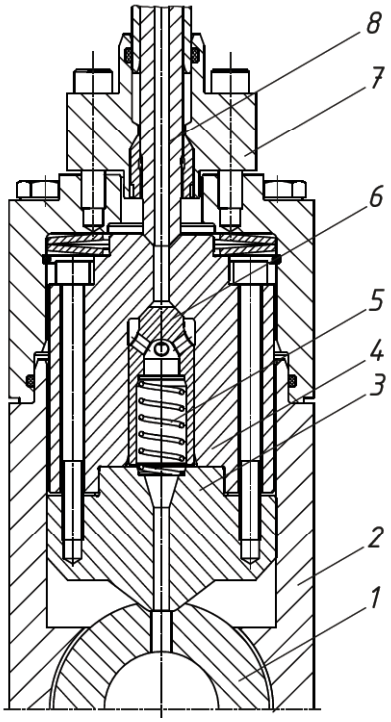


Figure 5.83 – Check valve of the X-35 series engine from WinGD: 1 – fuel accumulator; 2 – valve assembly housing; 3 – lower valve cover; 4 – valve housing; 5 – spring; 6 – valve; connecting flange; 7 – high pressure tube (adapted from [57])

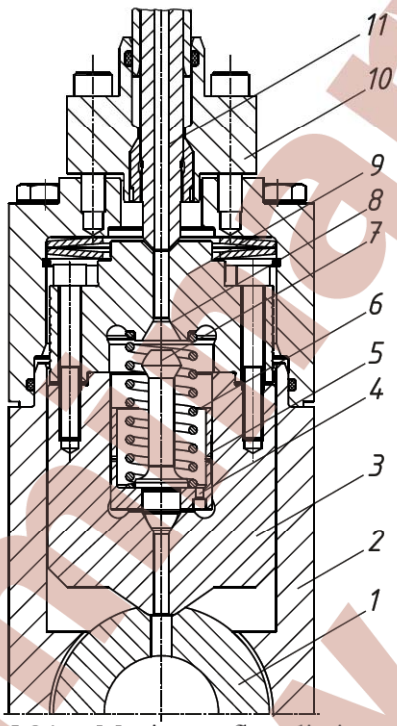


Figure 5.84 – Maximum flow limiter valve for the X-35 series engine from WinGD: 1 – fuel accumulator; 2 – valve assembly housing; 3 – valve housing; 4 – throttle hole; 5 – Piston flow limiter; 6 – spring; 7 – valve shut-off cone; 8 – valve seat; 9 – upper valve cover; 10 – connecting flange; 11 – high pressure tube (adapted from [57])

To monitor the pressure on the accumulators, several sensors are installed, which makes it possible to minimize deviations caused by possible pulsations in the accumulator, thereby ensuring objective pressure control.

The Common Rail system injector (Fig. 5.85, 5.86) is constantly under pressure equal to the pressure in the accumulator, which acts on the lower edge of the fuel injector shut-off valve. The valve opening pressure is determined by the spring pre-tension force.

Additionally, the upper end surface of the shut-off valve is subject to a force from the fuel pressure in the gas-locking chamber formed by the guide bushing and the upper end of the shut-off valve rod. Fuel enters the hydraulic shut-off chamber through the throttle housing and creates a force that holds the valve closed.

When voltage is applied to the control valve solenoid, the armature, under the influence of a magnetic field, overcomes the spring force. The control valve needle rises and connects the hydraulic shut-off chamber to the drain line (approximately 0.4 MPa). As a result, the shut-off valve rises and injection begins.

To stop injection, the solenoid is de-energized, and the control valve needle closes the drain from the hydraulic shut-off chamber. The fuel pressure in the chamber increases to the pressure in the accumulator, and the force created tightly presses the shut-off valve to its seat in the nozzle housing.

In the injectors used by MAN (Fig. 5.85), if the shut-off valve fails, the built-in flow limiter will operate, which will prevent the uncontrolled flow of fuel into the working cylinder. The volume of the flow limiter is 130% of the volume of fuel required to operate the engine at full load. The internal cavity of the injector acts as an additional pressure accumulator, which allows reducing the

For preliminary review

pressure drop during the fuel injection process. To pump heated fuel through the fuel injector when the engine is not running, a automatic fuel recirculation valve is built into the top cover of the housing. It opens under the action of a spring when the fuel pressure drops below 1.5 MPa.

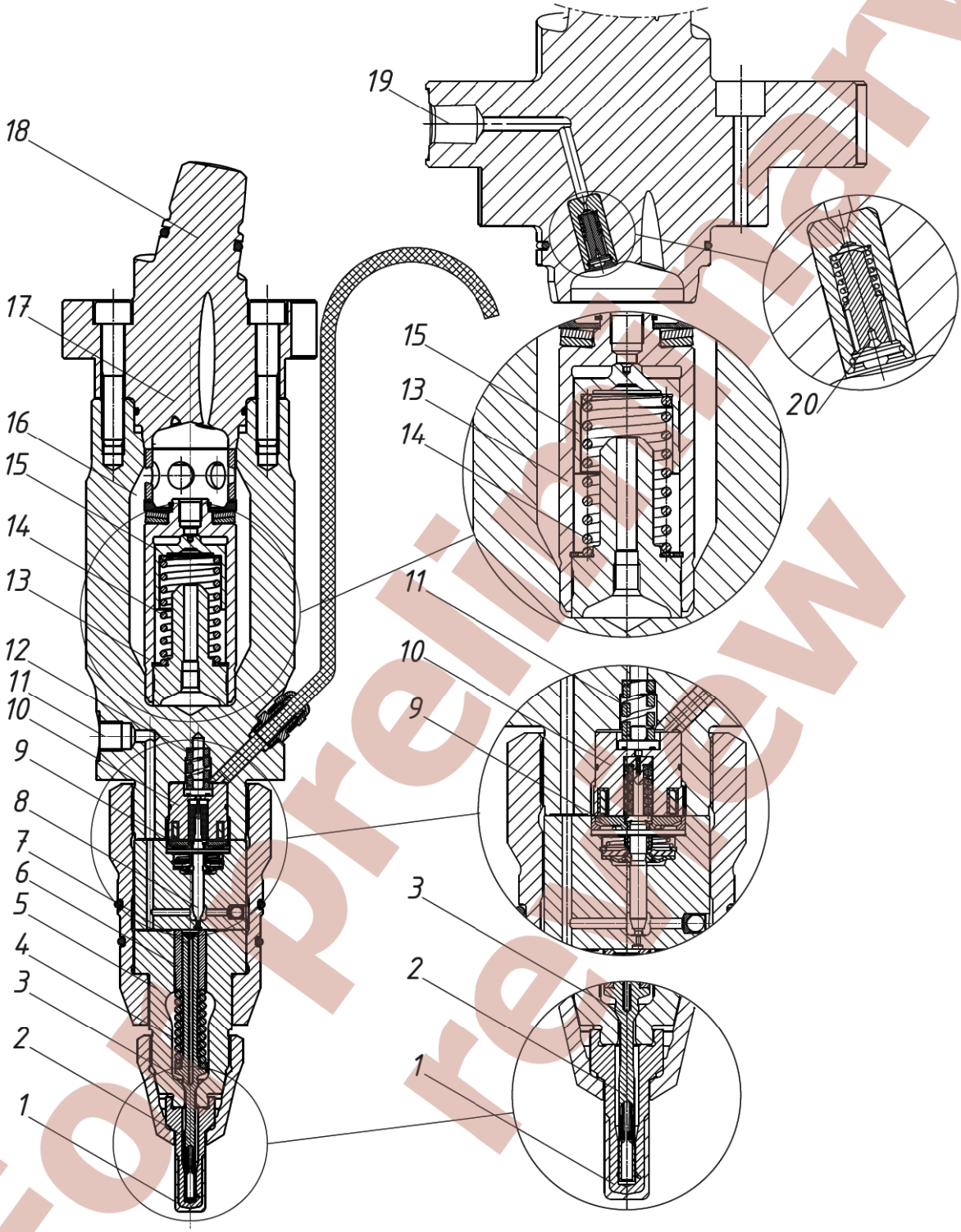


Figure 5.85 – Fuel injector with electrohydraulic control used in low-speed MAN engines equipped with a accumulator fuel injection system: 1 – nozzle tip; 2 – spool valve of the nozzle tip; 3 – nozzle shut-off valve; 4 – shut-off valve spring; 5 – nozzle housing; 6 – guide sleeve of the hydraulic locking chamber; 7 – hydraulic locking piston; 8 – unloading valve of the hydraulic shut-off chamber; 9 – solenoid armature; 10 – solenoid; 11 – channel for draining fuel from the hydraulic locking chamber; 12 – fuel injector housing; 13 – valve bushing for limiting the maximum flow; 14 – spring; 15 – maximum flow limitation valve; 16 – accumulator cavity; 17 – cover; 18 – fuel supply fitting; 19 – channel for fuel removal during circulation; 20 – circulation valve (adapted from [58])

For preliminary review

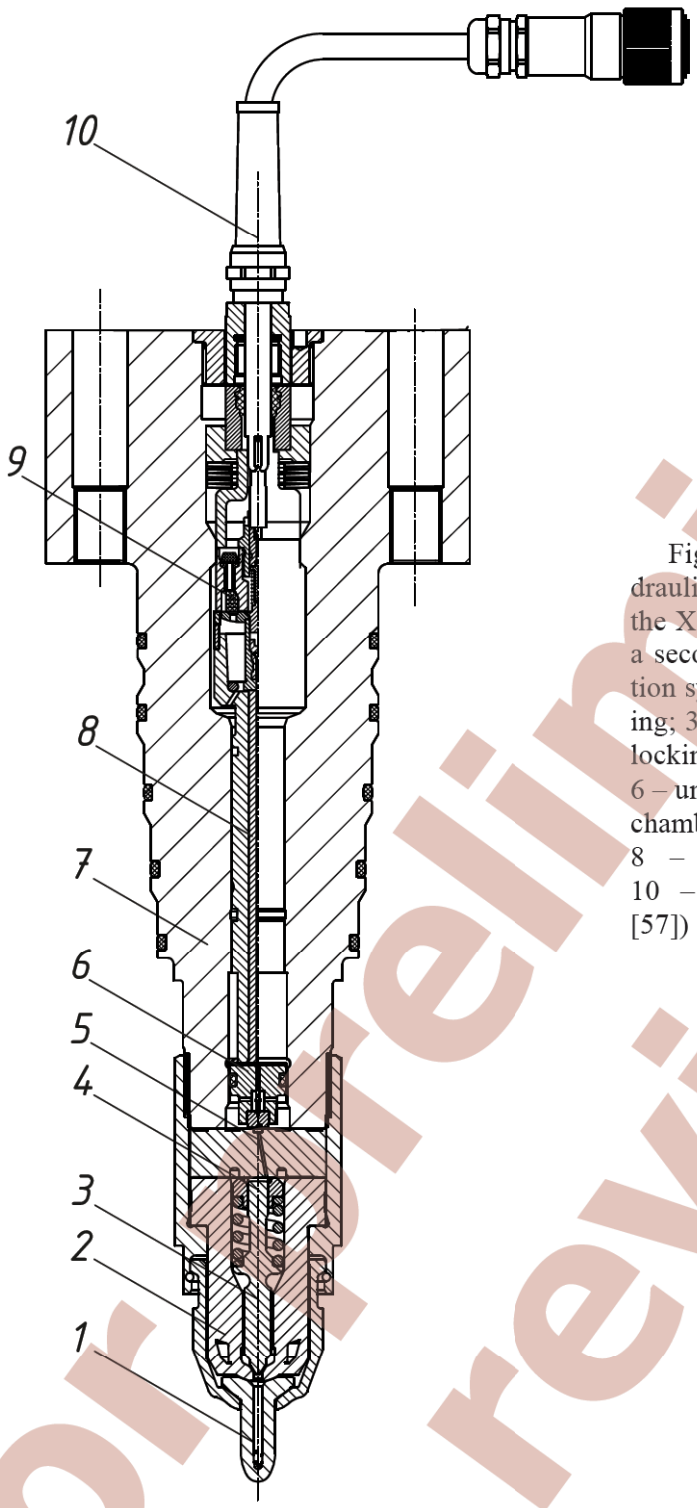


Figure 5.86 – An injector with electrohydraulic control used in low-speed engines of the X-35 series from WinGD equipped with a second generation accumulator fuel injection system: 1 – nozzle tip; 2 – nozzle housing; 3 – nozzle shut-off valve; 4 – hydraulic locking piston; 5 – throttle channel; 6 – unloading valve of the hydraulic shut-off chamber; 7 – fuel injector housing; 8 – pusher rod; 9 – solenoid valve; 10 – electrical connector (adapted from [57])

Power is supplied to the injector solenoid coil through the amplifier module (Common Rail Injection System Driver – CRISD). The CRISD module is powered and controlled by the universal controller (Multi-Purpose Controller – MPC). When an injection command is given, a relatively large current is required to raise the control valve actuator armature. After opening, to keep the valve open, the current density supplied to the solenoid is reduced. The supply voltage during opening is 100V DC, and the power control is carried out using pulse width modulation method.

5.7.3 Accumulator injection systems for medium-speed diesel engines

In recent years, leading manufacturers of marine medium-speed diesel engines have undertaken significant research and development work to develop accumulator injection systems for medium-speed engines. As a result of this work, a fundamentally new class of diesel engines has appeared

for preliminary review

on the market, which can meet existing and future standards for the content of harmful substances in exhaust gases. The bulk of the engines were created on the basis of already existing models that have proven themselves in operation. At the same time, MAN launched a fundamentally new engine of the L(W) 32/44CR series, which was originally designed with a common rail injection system. Subsequently, the injection system developed for this engine was used on other models produced by this company.

When developing common rail systems for medium-speed diesel engines, almost all manufacturers abandoned the use of a single large-volume accumulator, replacing it with several accumulator modules that provide fuel supply, as a rule, to two or three engine cylinders. This decision is explained by the following considerations:

- medium-speed engines use a wider range of fuels, which require heating temperatures in the range of 25…150°C to achieve a given viscosity. This in turn causes significant differences in the linear thermal expansion of the accumulator;
- in a long accumulator, it is inevitable that there will be radial drillings for installing connecting fittings and pipelines that carry fuel to individual engine cylinders. These drillings are concentrators of significant stresses that weaken the structure. The magnitude of these voltages is proportional to the pressure in the accumulator and its internal diameter. Reducing the harmful effects of these stresses, for example, by increasing the wall thickness, sometimes leads to an unjustified increase in the dimensions and weight of the fuel supply system as a whole;
- reducing the diameter of the accumulator leads to a reduction in its volume, which, provided that the cyclic supplys are sufficiently large, makes it difficult to achieve the same injection conditions in different cylinders, especially if fuel under high pressure is supplied to the accumulator from one side. In addition, excessive pressure fluctuations may occur in the accumulator, disrupting the fuel supply;
- the use of small-volume batteries makes it easier to integrate them into an existing engine design, using standard mating surfaces for their fastening.

It should be noted that the use of several accumulator sections does not fully correspond to the name of the Common Rail systems however; this name has traditionally been applied to all fuel systems of accumulator-type diesel engines.

Next, we will consider the design features of accumulator injection systems of some leading manufacturers of marine diesel engines.

5.7.3.1 The MAN accumulator fuel supply system is made according to a modular design, which allows it to be used on various engine models without significant changes in the design of individual elements. The system is designed to use a variety of fuels, including heavy residual fuel HFO with a viscosity of up to 700 cSt at 50°C. The design of the system allows the engine to operate on heavy fuels in all operating modes, including engine start and stop modes.

To drive the spool valves that control the fuel supply, fuel taken from the accumulator is used. High-speed valves that control fuel injection are located in the covers of the accumulator sections, which makes it possible to minimize the length of high-pressure pipelines and use standard injectors for fuel injection. The general hydraulic diagram of the system is shown in Fig. 5.87, and in Fig. 5.88 shows its general appearance.

The fuel prepared for use in the engine is supplied to the high-pressure pumps through throttle valves controlled by an electromagnetic drive.

Under the influence of a signal coming from the microprocessor control unit, the valves change the flow area of the throttle opening, thereby regulating the amount of fuel entering the space above the plunger of the pump, where it enters through the suction valve. From the pumps, fuel under pressure up to 160 MPa flows through the injection valves into the accumulators, on the end covers of which there are fuel injection control valve blocks. In addition to the valve block, connecting fittings for the high-pressure pipelines to the fuel injector and the connecting pipeline between the sections of the accumulator are installed on the end covers of the accumulators. The accumulator case is a thick-walled tube, in which there are no radial drillings, which ensures its maximum rigidity. The volume of the accumulators is selected in such a way as to ensure constant pressure in the

For preliminary review

system during operation without the use of gas or spring compensators.

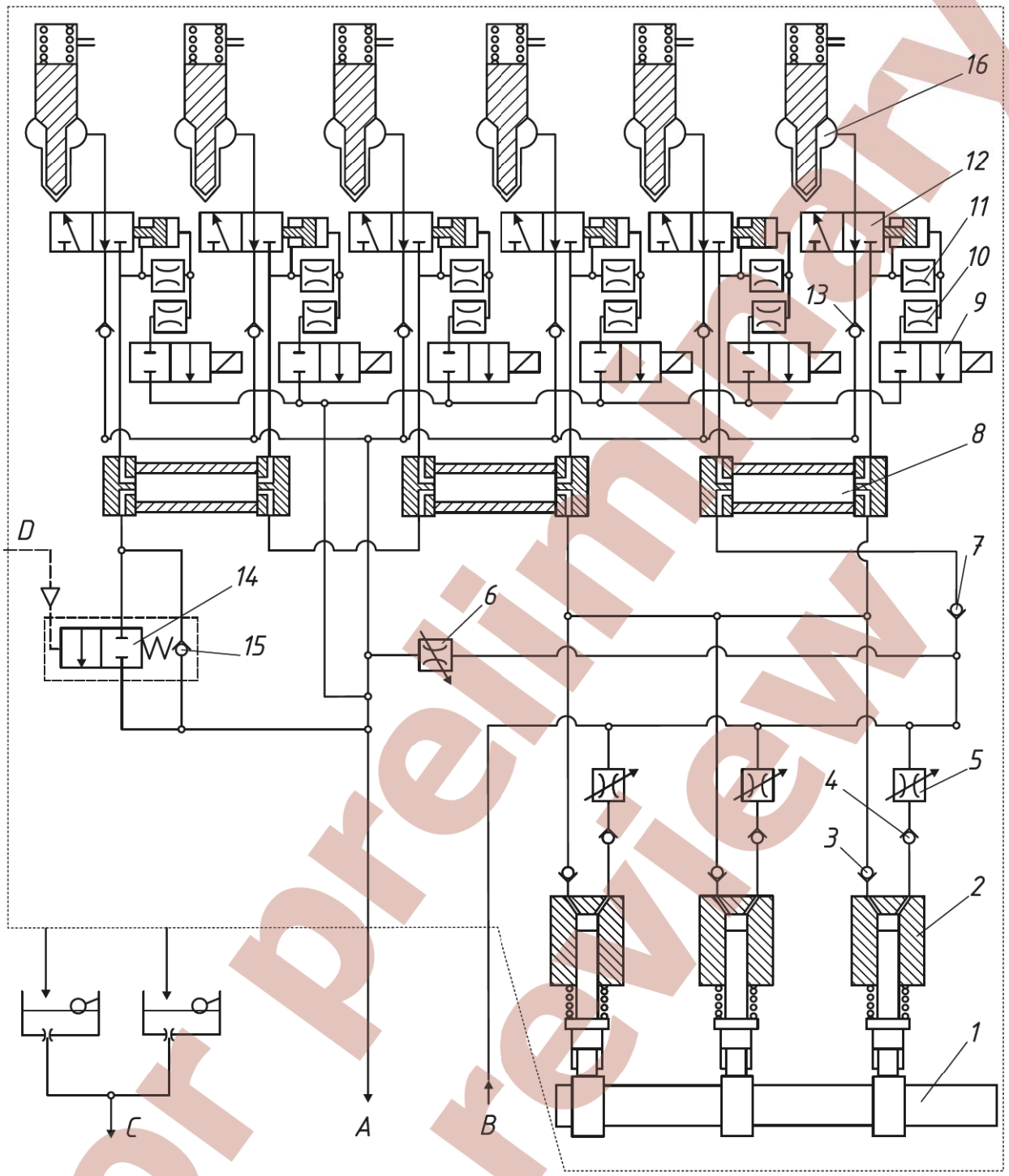


Figure 5.87 – Hydraulic diagram of the accumulator fuel system of MAN medium-speed engines: 1 – pump drive shaft; 2 – high pressure fuel pumps; 3 – delivery valve; 4 – suction valve; 5 – flow control throttle; 6 – throttle to maintain pressure difference; 7 – check valve for increasing flow when pumping the system; 8 – accumulator; 9 – starting valve with electromagnetic drive; 10 – starting valve throttle; 11 – throttle of the main supply control valve; 12 – main supply control valve; 13 – check valve of the drain cavity; 14 – system bleeding and emergency pressure relief valve; 15 – high pressure limiting valve; 16 – fuel injector. *A* – fuel supply; *B* – fuel drain; *C* – collection of leaks; *D* – control air supply (adapted from [59])

To minimize pressure fluctuations in the system, fuel supply to the batteries is provided by several pumps driven by a shaft on which three-jaw washers are installed.

The washers are wedged relative to each other at an angle that ensures an equal displacement of

For preliminary review

the supply phases between the individual pumps. Depending on the load-speed mode of the engine, due to throttle control, fuel pumps supply the system with as much fuel as is necessary to compensate for its consumption for injection and fuel supply control. Due to this, regardless of the operating mode, the pressure in the system is maintained constant.

The total supply of the pumps is selected in such a way that, if one of them is damaged, the others are able to provide the system with the necessary amount of fuel for the engine to operate at rated power.

Each accumulator section is equipped with two fuel control valve blocks. Accordingly, one accumulator section serves two engine cylinders. If there is an odd number of cylinders, an additional section with one valve block is installed.

Fuel is supplied from the accumulator to the injectors through the main injection control valve, which is hydraulically driven. The fuel taken from the accumulator is used as the working fluid for the hydraulic drive. The hydraulic drive of the main valve is controlled by a solenoid driven starting valve, which receives signals to open or close from the electronic engine control unit. To prevent continuous supply of fuel to the cylinder, in case of jamming of the main valve, maximum flow limiting valves are installed in the internal cavity of the accumulator, on the end covers, which automatically turn off the damaged control valve block.

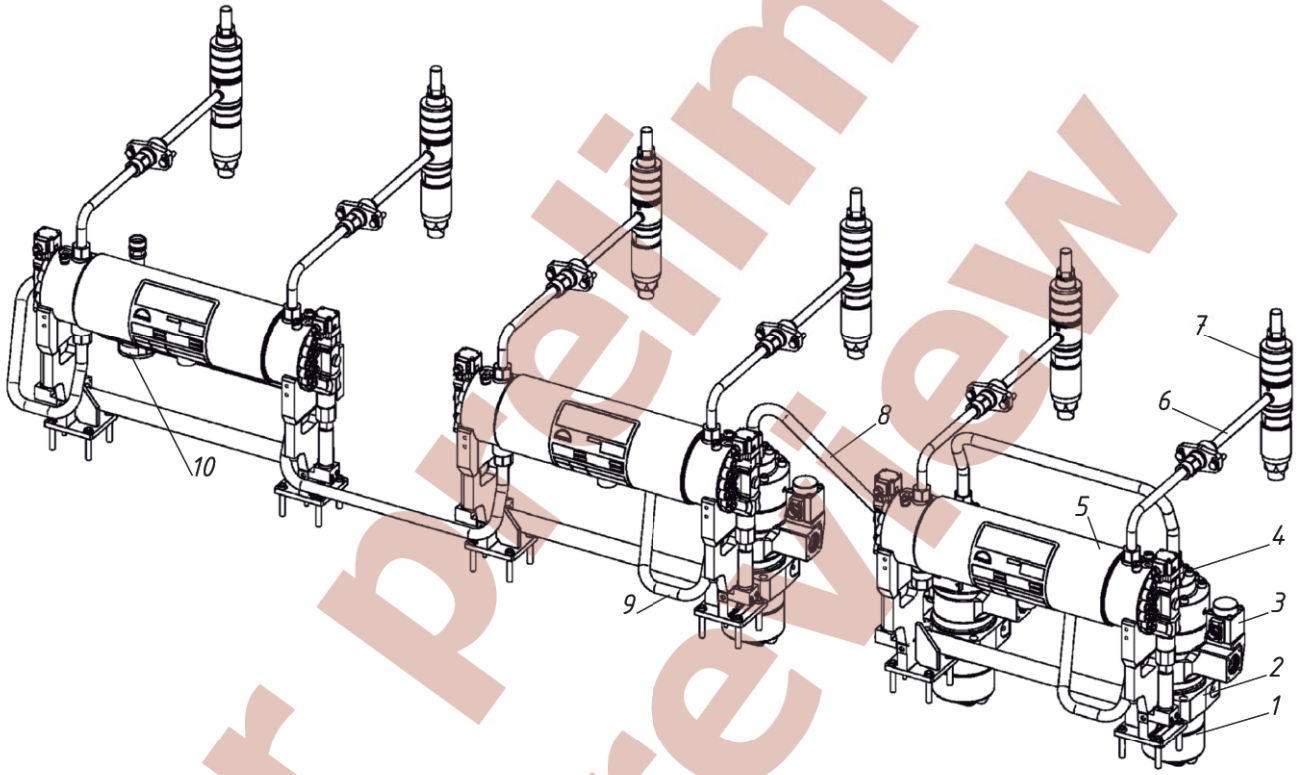


Figure 5.88 – General view of the accumulator fuel system of the 6L32/44 medium-speed engine from MAN: 1 – high pressure fuel pump; 2 – check valve; 3 – electrically controlled flow control throttle; 4 – supply control valve block; 5 – accumulator; 6 – high pressure line; 7 – fuel injector; 8 – high pressure connecting pipeline; 9 – fuel supply pipeline to the pumps; 10 – block of valves for bleeding and emergency pressure relief (adapted from [60])

To pump the system with heated fuel during parking or after a long period of inactivity with the engine supply systems turned off, as well as for emergency pressure relief in the batteries, a special valve is installed, actuated by compressed air from the control system. A pressure limiting valve is installed in the same block with the bleeding valve, which is activated when the pressure in the system exceeds the set value.

High-pressure fuel pumps used by MAN are designed according to the traditional design. Unlike injection pumps of volumetric injection systems, they do not have spool edges for supply control (Fig. 5.89), and there are no filler or cut-off holes in the bushing. The pump cover has a suc-

For preliminary review

tion valve designed to fill the space above the plunger with fuel.

To control the performance of the pumps, regulation is applied by throttling the fuel at the inlet. This method is not highly accurate, but in this case it is not necessary, since the cyclic portion is measured not by the pump itself, but by the control valve, which receives signals from the electronic controller. The microprocessor controller also controls the filling throttle valves, which have an electromagnetic drive and are mounted directly on the pump housing (Fig. 5.89).

In addition to the suction valve, a delivery valve is installed in the pump cover, separating the above-plunger cavity and the high-pressure line in the absence of discharge.

To supply the engine with fuel under high pressure, several pump sections are installed, which are driven from a cam shaft with three-cam washers installed on it (Fig. 5.90).

The cam shaft is driven through a gear transmission from the engine crankshaft. The number of pumps installed on one engine depends on the number of its working cylinders.

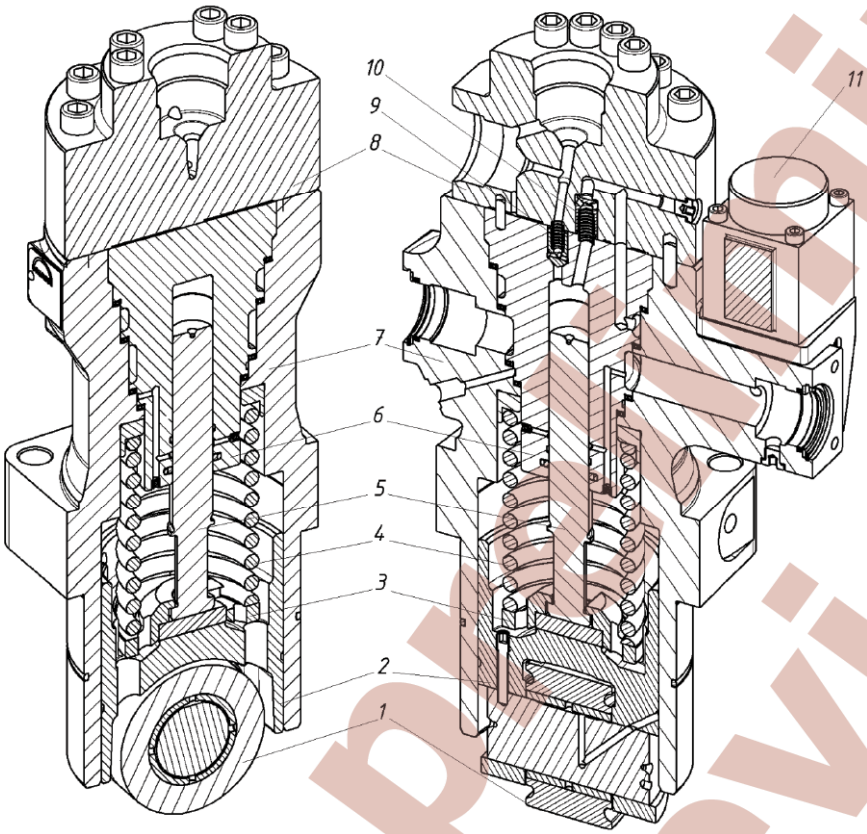


Figure 5.89 – High-pressure fuel pump for MAN series 32/44CR engines: 1 – pusher roller; 2 – pusher; 3 – lower plate of the return spring; 4 – return spring; 5 – plunger; 6 – plunger bushing; 7 – housing; 8 – cover; 9 – delivery valve; 10 – suction valve; 11 – flow control throttle valve with electromagnetic drive (adapted from [60])

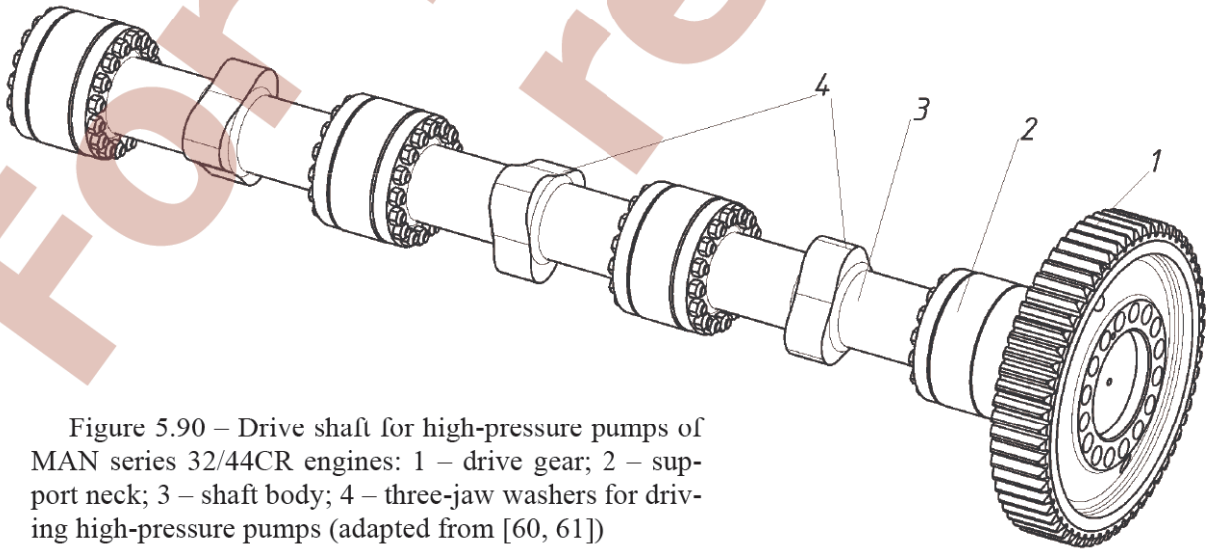


Figure 5.90 – Drive shaft for high-pressure pumps of MAN series 32/44CR engines: 1 – drive gear; 2 – support neck; 3 – shaft body; 4 – three-jaw washers for driving high-pressure pumps (adapted from [60, 61])

For preliminary review

The pressure accumulator is made in the form of a horizontally located thick-walled pipe, closed at the ends with covers attached to the housing with bolts (Fig. 5.91).

For greater safety, the accumulator case is placed in a protective metal casing, which protects against fuel spillage in the event of a accumulator rupture. A space is formed between the housing and the casing, which is used to collect leaks, which are drained from the accumulator through a fitting located in the lower part of the casing.

The leakage collection tube connects the fitting on the casing to the collection box, in which a capacitive leakage monitoring sensor is installed (Fig. 5.91).

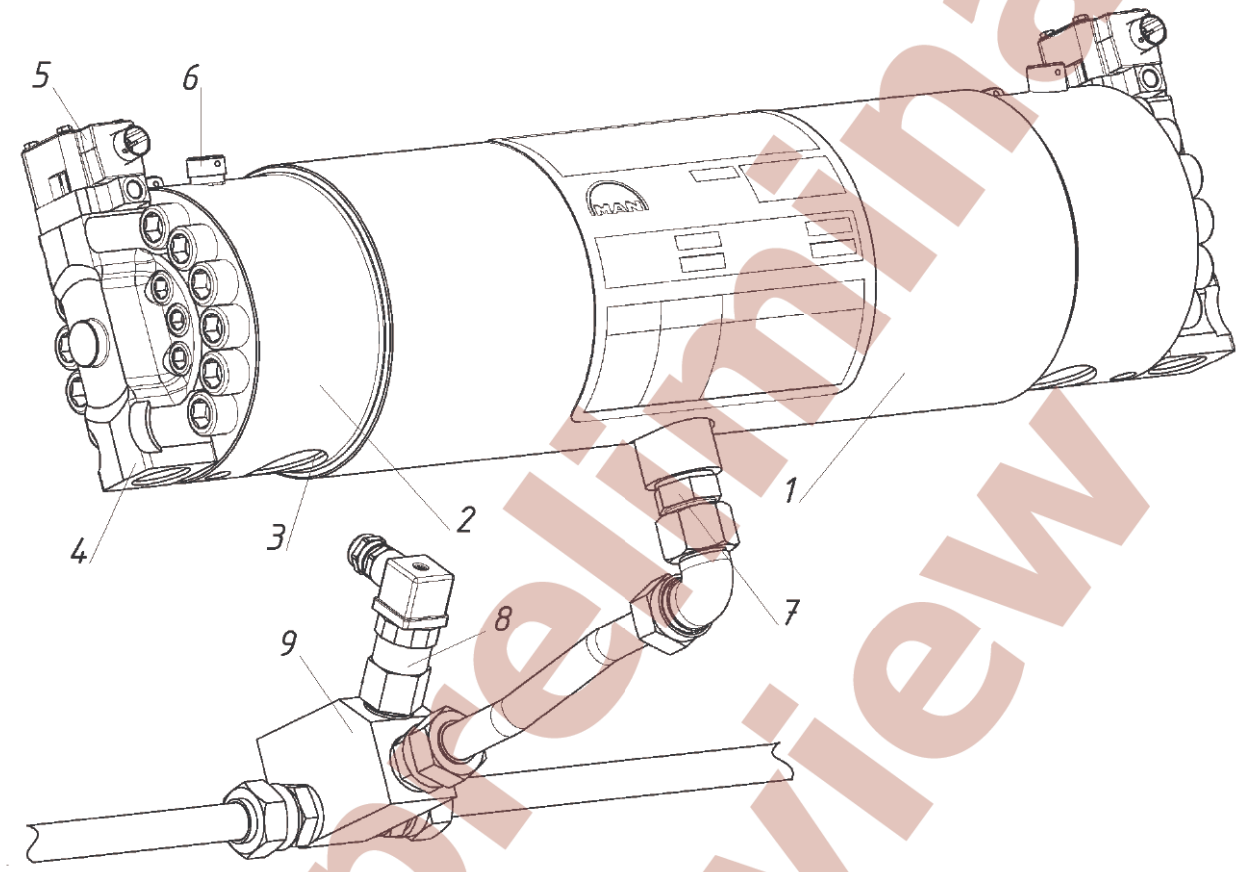


Figure 5.91 – General view of the accumulator for engines of the 32/44CR series from MAN: 1 – accumulator protective casing; 2 – end covers; 3 – connecting hole of the high pressure line; 4 – fuel supply control valve block; 5 – start valve solenoid; 6 – holes for connecting control sensors; 7 – leakage collection fitting; 8 – leakage monitoring sensor; 9 – leakage collection box (adapted from [60, 61])

If the established norm is exceeded, the sensor sends a signal to the controller. The latter signals the presence of a malfunction, and if there is a threat of an accident, it stops the engine.

All connecting fittings of the accumulator are located on the side covers, to which a maximum flow limiter valve is attached from the side of the accumulator cavity, and from the outside – a block of fuel supply control valves.

The general arrangement of elements of the fuel supply control unit is shown in Fig. 5.92.

The maximum flow limiter valve serves to prevent uncontrolled fuel injection in the event of failure of the main fuel control valve (Fig. 5.92).

The valve consists of a hydraulic cylinder attached to the accumulator cover using a threaded fitting. The fitting is drilled to supply fuel to the main control valve. The inner edge of the hole is turned to a cone, forming a limit valve seat. The operating principle of the limiter valve is identical to the similar valve of the Common Rail system of low-speed engines, which we reviewed earlier.

The fuel control valve block includes two valves: a main valve with a hydraulic drive and a starter with an electromagnetic drive, which controls the main valve.

For preliminary review

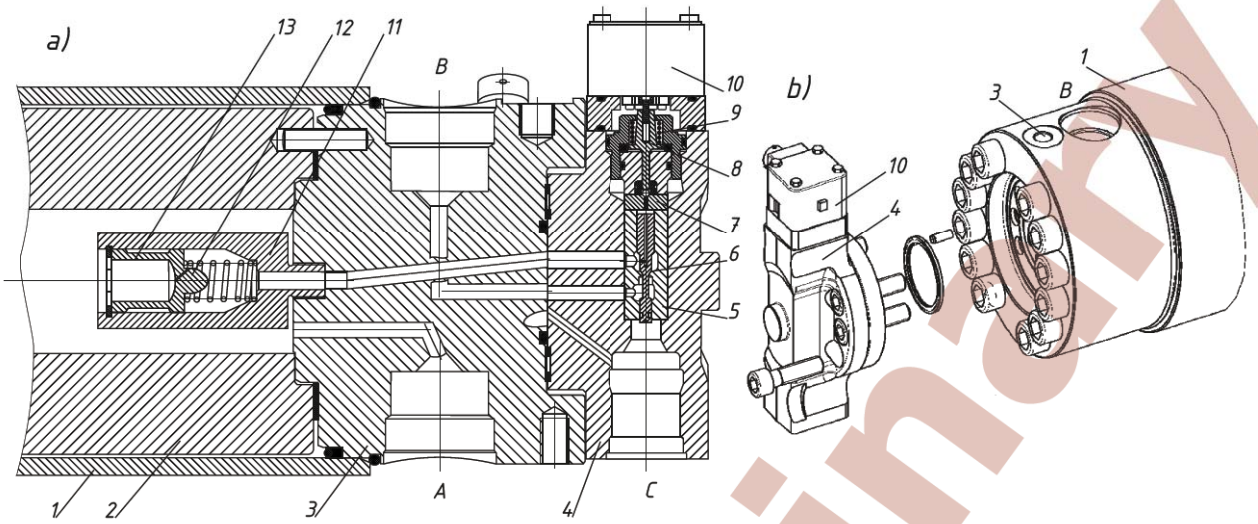


Figure 5.92 – Fuel supply control unit for engines of the 32/44CR series from MAN, general structure (*a*) and general view with the control valve removed (*b*): 1 – accumulator protective casing; 2 – accumulator housing; 3 – end cover; 4 – fuel supply control valve block; 5 – bushing of the main fuel control valve; 6 – main fuel supply control valve; 7 – seat of the ball starting valve; 8 – pressure rod of the starting valve; 9 – pressure spring; 10 – solenoid; 11 – hydraulic cylinder of the maximum fuel supply limiter valve; 12 – shut-off cone of the limiting valve; 13 – hydraulic piston of the limiter valve. *A* – high pressure fuel supply to the accumulator; *B* – fuel discharge to the injector; *C* – fuel dump to unload the high pressure line (adapted from [60, 61])

Starting valve communicates or disconnects the cavity of the hydraulic piston of the main valve drive with the drain line (Fig. 5.93). The main locking element is a steel ball, pressed to the seat by a spring through a rod, made as a continuation of the solenoid armature.

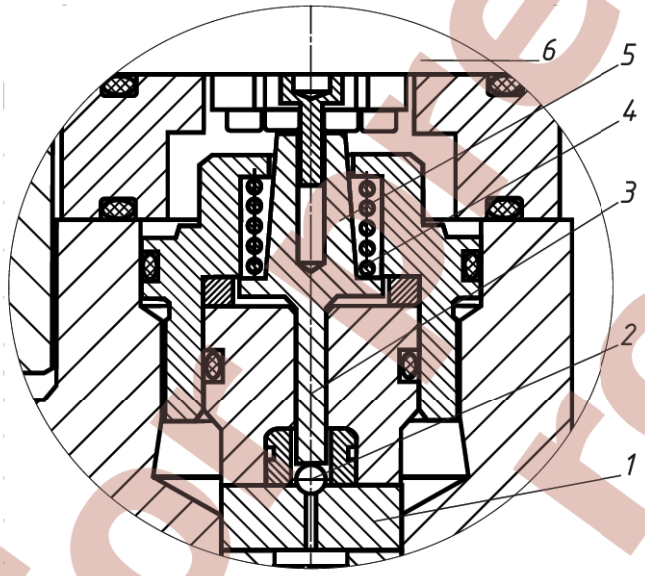


Figure 5.93 – Start valve of the fuel supply control unit: 1 – valve seat; 2 – ball; 3 – rod; 4 – pressure spring; 5 – solenoid armature; 6 – solenoid (adapted from [60, 61])

In the absence of a signal from the electronic control unit, a spring through the rod presses the ball to the seat, disconnecting the cavity of the hydraulic drive of the main supply control valve from the drain line.

When voltage is applied from the control unit to the solenoid, the armature, overcoming the force of the spring, rises upward, releasing the ball. Under the influence of pressure in the cavity of the hydraulic drive, the ball rises and releases fuel from this cavity into the drain line.

To ensure smooth movement of the main valve drive piston, the channel connecting the cavities to each other acts as a throttle.

The main fuel control valve has two control elements: cone (to control injection) and spool (to control fuel cut-off). To drive the valve, a hydraulic piston is used, made as one piece with the rest

For preliminary review

of the valve elements.

The general diagram and operating procedure of the main valve are shown in Fig. 5.94.

In the absence of a control signal from the electronic control unit, the start valve is in the closed position (Fig. 5.94 *a*), fuel enters the cavity of the main valve, from where part of it flows through the throttle channel into the cavity above the drive piston. Thus, the same pressure is established in both cavities, however, given that the area of the piston on top is much larger than on the side of the valve cavity, a difference in forces arises, pressing the main valve cone to the seat. In this case, fuel does not flow to the injector, there is no injection.

In the position when the main valve is closed, the edge of the shut-off control spool is below the shut-off edge located in the valve housing, connecting the fuel injector cavity with the drain line (Fig. 5.94 *a*).

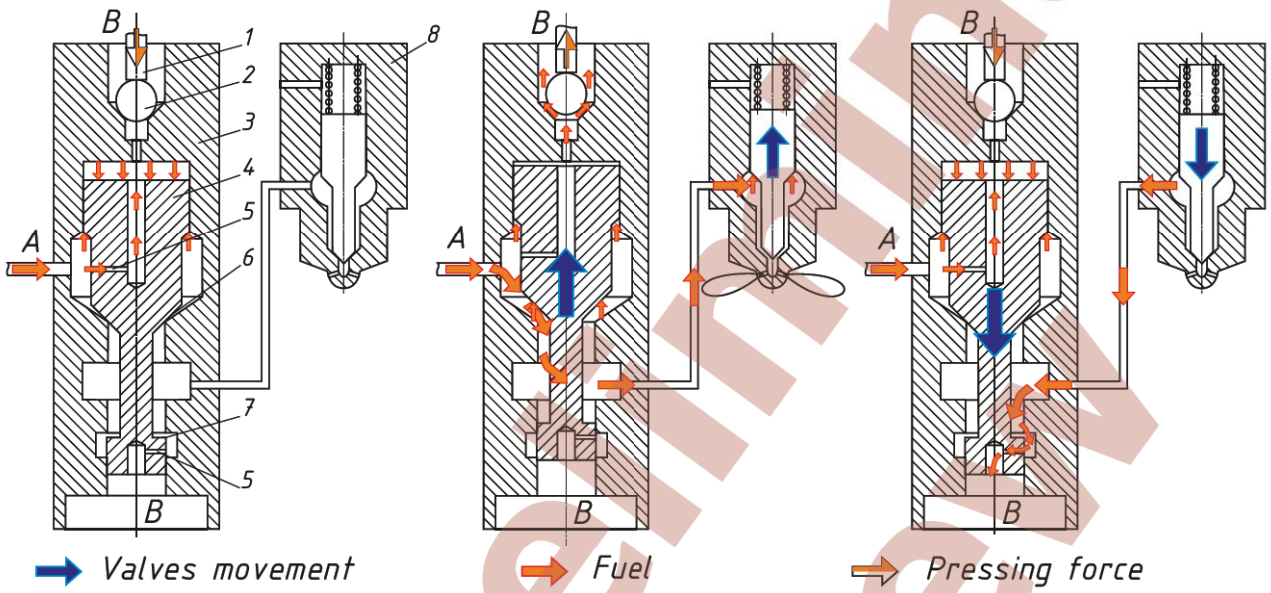


Figure 5.94 – Diagram of the operation of the main valve of the accumulator injection system of MAN series 32/44CR engines: *a* – the main valve is closed, there is no injection; *b* – the starting valve is open, the main valve is open, fuel is injected; *c* – the start valve is closed, the main valve is closed, the shut-off edge relieves pressure from the discharge pipeline. 1 – starting valve rod; 2 – ball; 3 – main valve housing; 4 – main valve drive piston; 5 – throttle hole; 6 – main valve; 7 – shut-off spool valve; 8 – fuel injector. *A* – fuel supply from the accumulator; *B* – drain cavity (adapted from [59])

To maintain some excess pressure in the high-pressure line, the drain cavity is equipped with a check valve, which simultaneously serves as a pressure limiting valve (not shown in the figure).

When a control signal is received from the control unit, the starting valve opens, as a result of which the pressure above the actuator piston is significantly reduced. The force acting on the piston from above becomes less than the force acting on it from below. Under the influence of the difference in forces, the piston rises upward, displacing fuel from the above-piston cavity (Fig. 5.94 *b*). As a result of movement, the cone valve connects the accumulator cavity with the high pressure line, and the spool edge disconnects it from the drain cavity. The fuel flows to the injector, which, after reaching the opening pressure of the needle valve, begins fuel injection (Fig. 5.94 *b*).

When the start valve closes, fuel fills the space above the piston through the throttle passage, moving the piston down, which causes the supply control cone valve to close. At the same time, the edge of the spool connects the high pressure line with the drain cavity, the pressure in front of the fuel injector drops sharply, fuel injection into the combustion chamber stops (Fig. 5.94 *c*).

The check valve separates the cavity for draining fuel from the high pressure line and the cavity for low pressure. It also serves as a pressure limiter in the drain cavity, maintaining some excess pressure in the discharge line (Fig. 5.95).

This pressure, required to reduce the time to reach the injection start pressure, increases the per-

For preliminary review

formance of the system as a whole.

In addition, the valve prevents the occurrence of wave processes in the pressure line. When the pressure wave reaches the valve, it opens and transfers part of the fuel into the damper cavity, in a large volume of which the wave is dissipated.

A ball valve loaded with a spring is used as a shut-off element. To adjust the pressure in the drain cavity, adjusting washers are placed under the spring.

The bleeding and emergency pressure relief valve is made in one block with the maximum pressure limiting valve (Fig. 5.96).

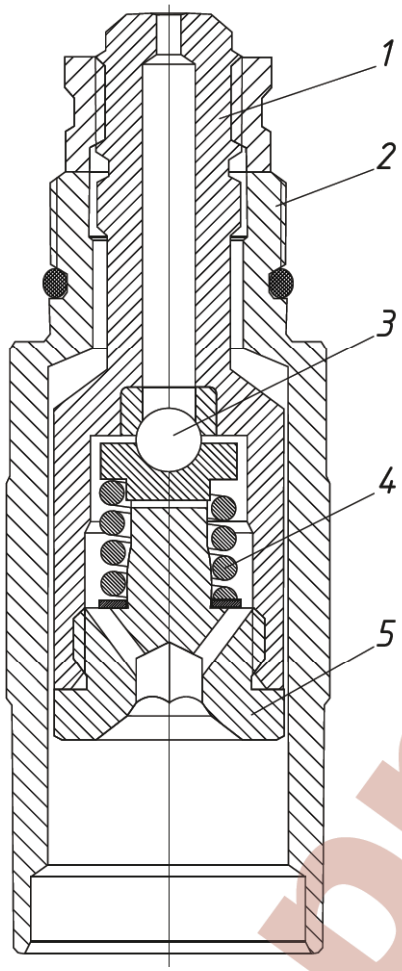


Figure 5.95 – Check valve of the fuel drain system: 1 – valve housing; 2 – outer housing; 3 – ball valve; 4 – pressure spring; 5 – screw plug (adapted from [60])

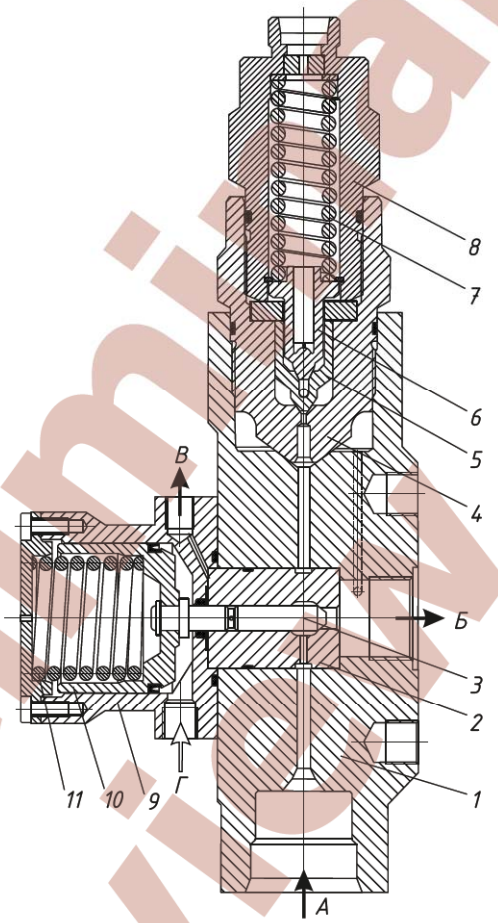


Figure 5.96 – Bleeding and emergency pressure relief valve and maximum pressure limitation valve: 1 – valve block housing; 2 – bleeding valve housing; 3 – bleed valve stem with shut-off cone; 4 – housing of the maximum pressure limiting valve; 5 – plug-in valve-limiter seat; 6 – shut-off cone of the limiting valve; 7 – loading spring; 8 – threaded plug; 9 – pneumatic cylinder for pumping valve drive; 10 – piston of the pumping valve drive; 11 – loading spring; *A* – fuel supply from the accumulator; *B* – drain line; *C* – collection of leaks; *D* – compressed air from the control system (adapted from [60, 61])

The valve is driven by a pneumatic piston, to which air is supplied from the control system.

The valve block is installed in the system at the outlet of the accumulator furthest from the pumps.

When the valve opens, the high-pressure cavity is connected to the drain line and the pressure in the system is released.

A valve is used to pump the fuel system with heated fuel in preparation for starting the engine on heavy fuel.

Opening the valve causes the pressure in the system to become lower than in the supply line. Under the influence of a pressure difference, the heated fuel is divided into two streams. The first flow through the open throttle control valves of the pumps, through the suction valves fills the fuel

For preliminary review

pumps and then enters the accumulator closest to the pumps. The second flow, through the automatically opening valve to increase the flow rate (position 7 in Fig. 5.97), directly enters the same accumulator. Here both flows merge and then pass through the remaining batteries, filling them with hot fuel. To ensure separation of the main flow, the required pressure difference is created using an adjustable throttle installed in the system (position 6 in Fig. 5.97).

In this way, the fuel system is pumped, as a result of which cold fuel is replaced with hot one.

A needle-type maximum pressure limiting valve with spring loading is used to relieve pressure if it rises above the established norm. This allows you to avoid excessive loads on the elements of the fuel system and protect them from destruction.

The electronic engine control unit is a microprocessor controller with redundancy of all basic control and emergency protection functions. Fuel system control is one of the functions of the safety and engine management system, called S_aCoS_{one} (Safety and Control System on Engine). This system receives information from sensors of the crankshaft position, rotation speed, load, boost pressure, pressure, fuel temperature, etc.

Based on the data received, the controller, using an algorithm installed in the form of software, calculates the values of fuel pressure in the accumulator, the phases of the beginning and end of fuel supply, and the shape of the supply law curve required for the current engine operating conditions.

In the event of a failure of the main control module, the backup one is automatically switched on. Both modules are interconnected by one system bus, through which they periodically exchange information. The transition time from one module to another does not exceed one millisecond.

Careful selection and calibration of all system elements allowed us to achieve significant success in organizing the engine's operating processe. In Fig. 5.97 it can be seen that the pressure in front of the fuel injector very quickly increases to a maximum and changes little throughout the injection.

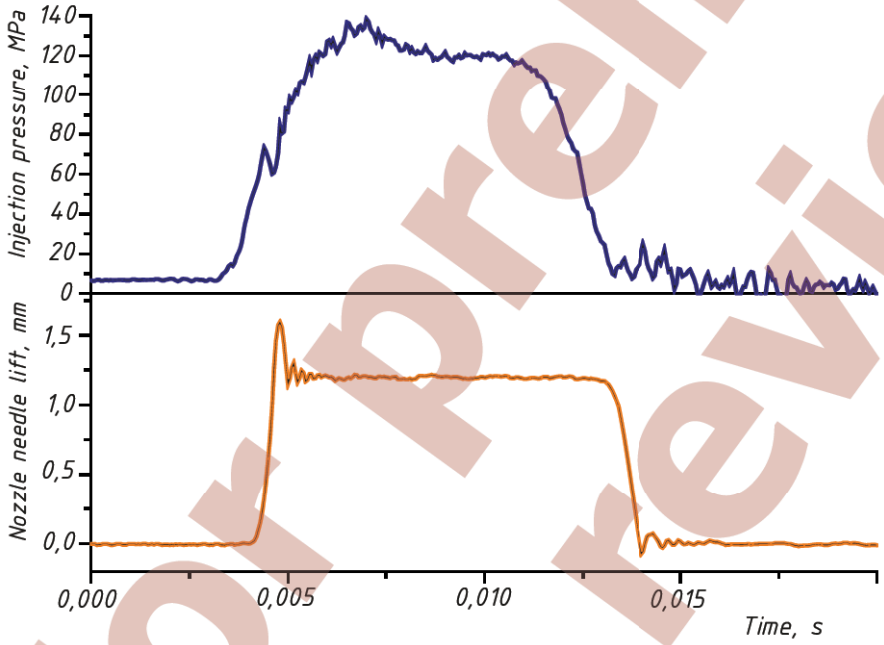


Figure 5.97 – Oscillograms of changes in fuel pressure in front of the injector and the injector needle lift curve (adapted from [62])

Bench tests of the fuel system have shown that it has a significant reserve for further improvement of fuel supply. The selection of individual parameters and control algorithms allows not only to change the beginning and end of fuel supply, but also to influence the nature of the flow of the supply curve, including multi-stage fuel injection.

The company's specialists believe that the system they have developed is capable of meeting not only existing, but also future standards for limiting emissions of harmful substances.

Currently, engines of the 32/44, 32/40, 48/60 series are equipped with this system.

5.7.3.2 The Wärtsilä accumulator fuel supply system consists of separate modules, which makes it more universal and allows it to be installed not only on newly produced engines, but also used to modernize diesel engines already in operation. The latter is achieved by unifying the main

For preliminary review

elements of the accumulator system with elements of the fuel supply systems already used on engines. The system is designed to operate on a variety of fuels, including heavy residual fuel such as HFO.

When developing the system, the company tried to minimally change the existing design of the engines, which largely determines the selected layout. In addition, when developing the fuel system, it was necessary to take into account the protective effect of numerous patents, which also influenced a number of design decisions.

The general diagram of the Wärtsilä accumulator fuel system is shown in Fig. 5.98. A general view of the main elements of the system is shown in Fig. 5.99.

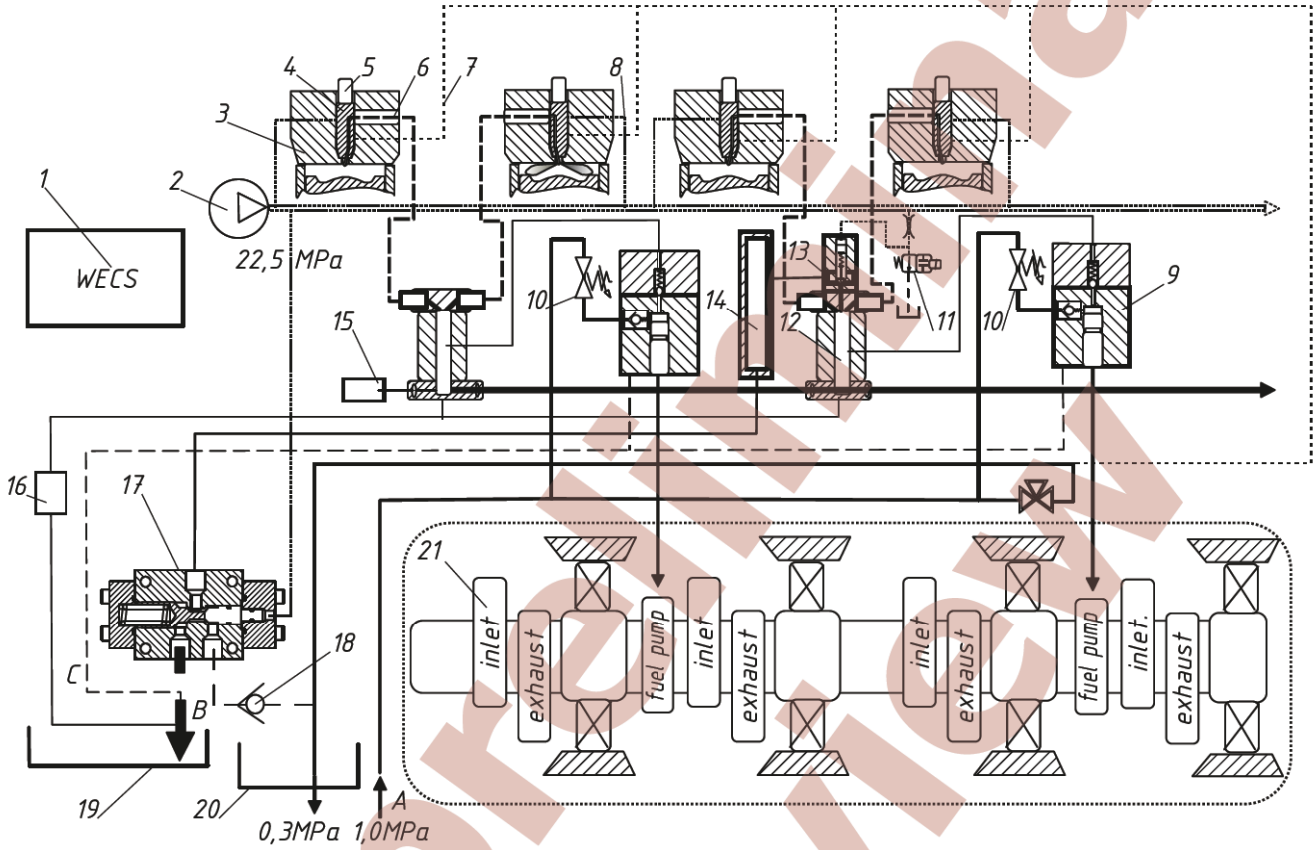


Figure 5.98 – Hydraulic diagram of the accumulator fuel system of medium-speed engines from Wärtsilä: 1 – electronic control unit; 2 – control oil pump; 3 – cylinder cover; 4 – fuel injector; 5 – fuel control solenoid; 6 – high pressure line; 7 – control oil drain line; 8 – control oil supply line; 9 – high pressure fuel pump; 10 – flow control throttle valve; 11 – control valve for the pressure regulator-limiter in the accumulator; 12 – accumulator; 13 – pressure regulator-limiter in the accumulator; 14 – buffer cavity; 15 – pressure sensor; 16 – leakage sensor; 17 – bleeding and emergency relief valve; 18 – check valve; 19 – leakage collection tank; 20 – mixing tank; 21 – camshaft; *A* – fuel supply; *B* – fuel drain; *C* – collection of leaks (adapted from [63])

From Figure 5.99 it can be seen that the company used a vertical arrangement of all elements of the system. This made the system more compact and its elements more accessible for maintenance. This arrangement is typical for traditional systems, therefore all elements of the accumulator system have the same connecting flanges as the traditional high-pressure pumps in which they are installed. This allowed the company to reduce the cost of implementing the system and ensure that its cost today is comparable to the cost of traditional fuel supply systems.

The main elements of the fuel system are:

- high pressure pumps;
- batteries equipped with a system of valves that control the pressure in the system;
- fuel injectors with electromagnetic supply control.

To drive various types of valves, a control oil system is provided, which is supplied to the system

For preliminary review

under a pressure of 22.5 MPa by axial plunger pumps. Before supply, the oil is pre-cleaned.

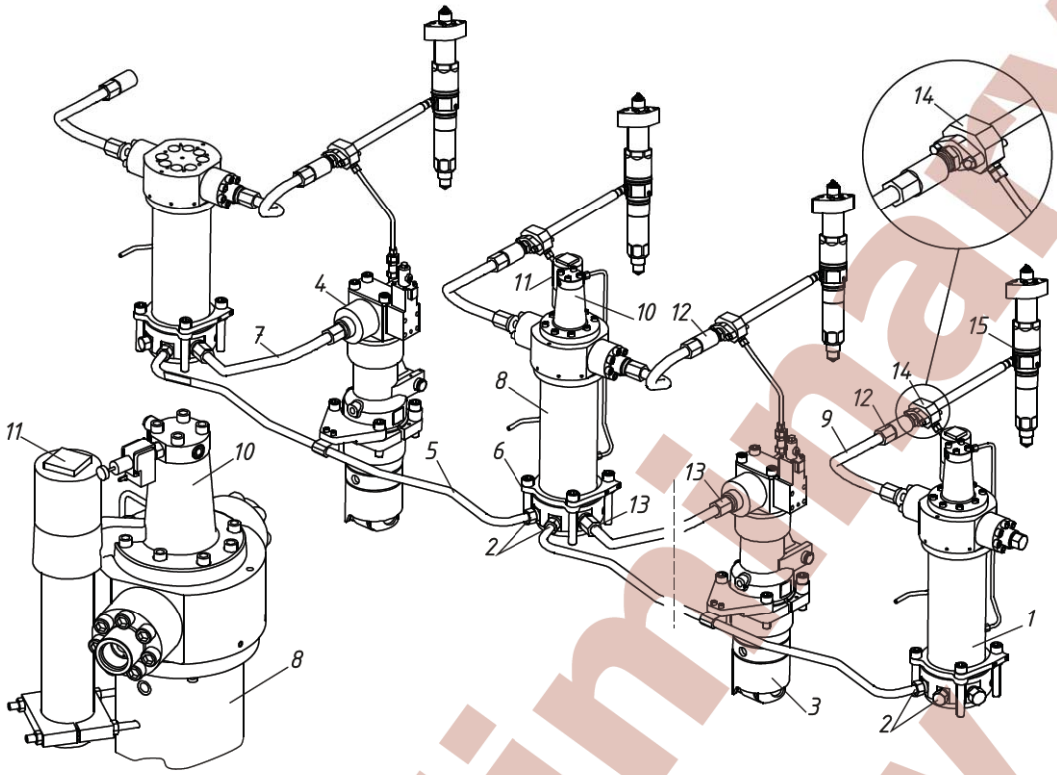


Figure 5.99 – General view of the elements of the accumulator fuel system of medium-speed engines from Wärtsilä 32 series: 1 – accumulator installation collar; 2 – fittings for connecting pipelines for supplying fuel under high pressure to the batteries; 3 – high pressure fuel pump; 4 – mounting flange of the high pressure pipeline to the fuel pump; 5 – high pressure pipeline connecting the batteries; 6 – accumulator connecting flange; 7 – high pressure pipeline from the pump to the accumulator; 8 – accumulator housing; 9 – pipelines for supplying fuel under high pressure to the injectors; 10 – pressure regulator-limiter in the accumulator; 11 – buffer cavity; 12 – injector adapter fittings; 13 – connecting fittings for the high pressure line; 14 – fuel injector fitting; 15 – injector with electromagnetic control of the injection process (adapted from [64])

The presence of a separate control oil system is dictated by the following considerations:

- for reliable operation of precision pairs of control valves, very fine cleaning of the working fluid is necessary, which is easier to do for oil than for heavy fuel;

- using heavy fuel with a temperature of 150°C or higher as a working fluid can lead to overheating and failure of the solenoid valves located in the fuel injector. Considering that these valves are already located in the cylinder cover, where they are exposed to high temperatures, the likelihood of their failure as a result of overheating is quite high. The control oil, having a lower temperature, among other things, cools the valve.

high-pressure fuel pump is made according to the traditional design (Fig. 5.100) and is driven by a camshaft on which a two-cam washer is installed. This made it possible to reduce the number of pumps by half; one pump provides a fuel supply sufficient to operate two engine cylinders.

The performance of the pumps is selected in such a way that, if one of them fails, the others will provide a supply sufficient to operate the engine at 80% of the nominal power.

The fuel pump itself is structurally simpler than its traditional counterpart; the absence of cut-off edges has significantly increased its reliability and the pressure it develops (Fig. 5.100 *a*, *b*). In the fuel system under consideration, the operating pressure of the pump lies in the range of 90...150 MPa, depending on the engine operating mode.

To prevent the plunger from being subjected to lateral forces that occur when the pusher roller rolls onto the cam washer, the pump has a separation between the drive mechanism and the pump section. Thanks to this, only forces directed along its axis are transmitted to the plunger.

The pump supply is controlled using throttling valves with an electromagnetic drive

For preliminary review

(Fig. 5.100 *c*). The flow area at the pump inlet is changed by moving the control cone along the vertical axis. The throttling valve is controlled by an electronic unit, which, depending on the operating mode, changes the flow of the pumps and the pressure under which the fuel enters the system.

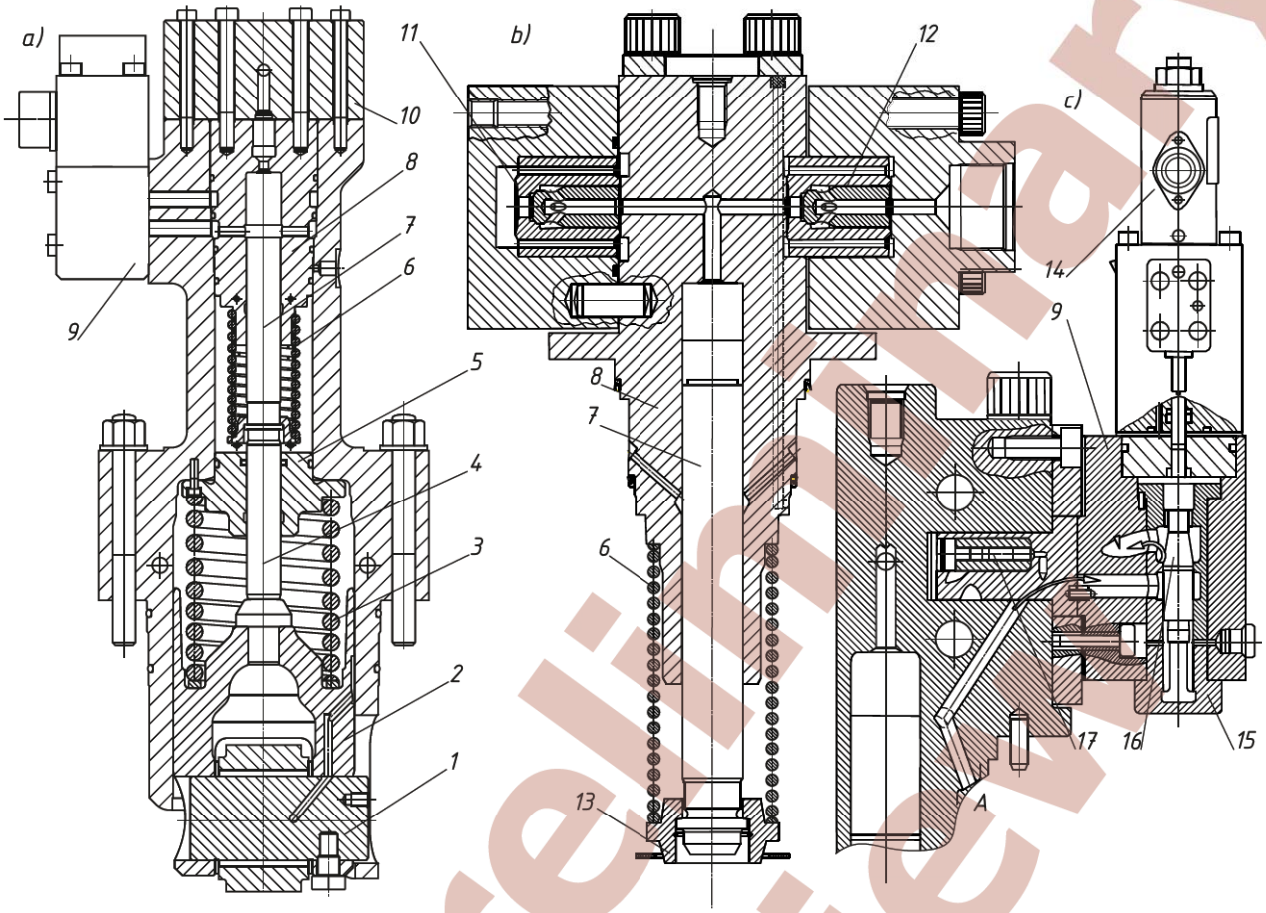


Figure 5.100 – High pressure fuel pump of the 32 series engine from Wärtsilä (*a*), pump section of the fuel pump (*b*) and throttle valve for controlling the fuel supply of 46 series engines from Wärtsilä (*c*): 1 – roller pusher axis; 2 – pusher; 3 – pusher return spring; 4 – pusher rod; 5 – dividing diaphragm with pusher rod guide; 6 – plunger return spring; 7 – plunger; 8 – plunger bushing; 9 – flow control throttle valve; 10 – pump cover; 11 – suction valve; 12 – delivery valve; 13 – lower plate of the plunger return spring; 14 – throttle valve drive solenoid; 15 – throttle valve housing; 16 – throttling cone of the supply control valve; 17 – safety valve; *A* – low pressure fuel supply (adapted from [64, 65])

The pressure accumulator is a thick-walled pipe made of high-quality steel, closed on both sides with covers that are attached to the housing with bolts (Fig. 5.101 *a*). All connecting fittings and various types of valves are concentrated on the accumulator covers. This makes it possible to increase the strength of the housing and ensure its reliable operation at sufficiently high pressures.

To maintain a given operating pressure in the fuel supply system, a pressure limiter regulator is installed on a number of batteries (Fig. 5.101 *b*).

On six-cylinder engines, one regulator is installed; on engines with a large number of cylinders, one regulator is installed per 5-6 cylinders.

Pressure limiter regulator (Fig. 5.101 *b*) is a multifunctional valve system that performs the following functions:

- together with the flow control throttle valve, maintains the specified pressure in the fuel system;
- protects the accumulator and fuel system from excessive pressure increases;
- ensures pumping of the system with hot fuel during parking or before starting the engine;
- prevents the occurrence of pressure waves in the fuel system.

The regulator is controlled by an electronic module through a throttling valve with an electro-

For preliminary review

magnetic drive.

Oil from the control oil system is used as the working fluid to control the valves.

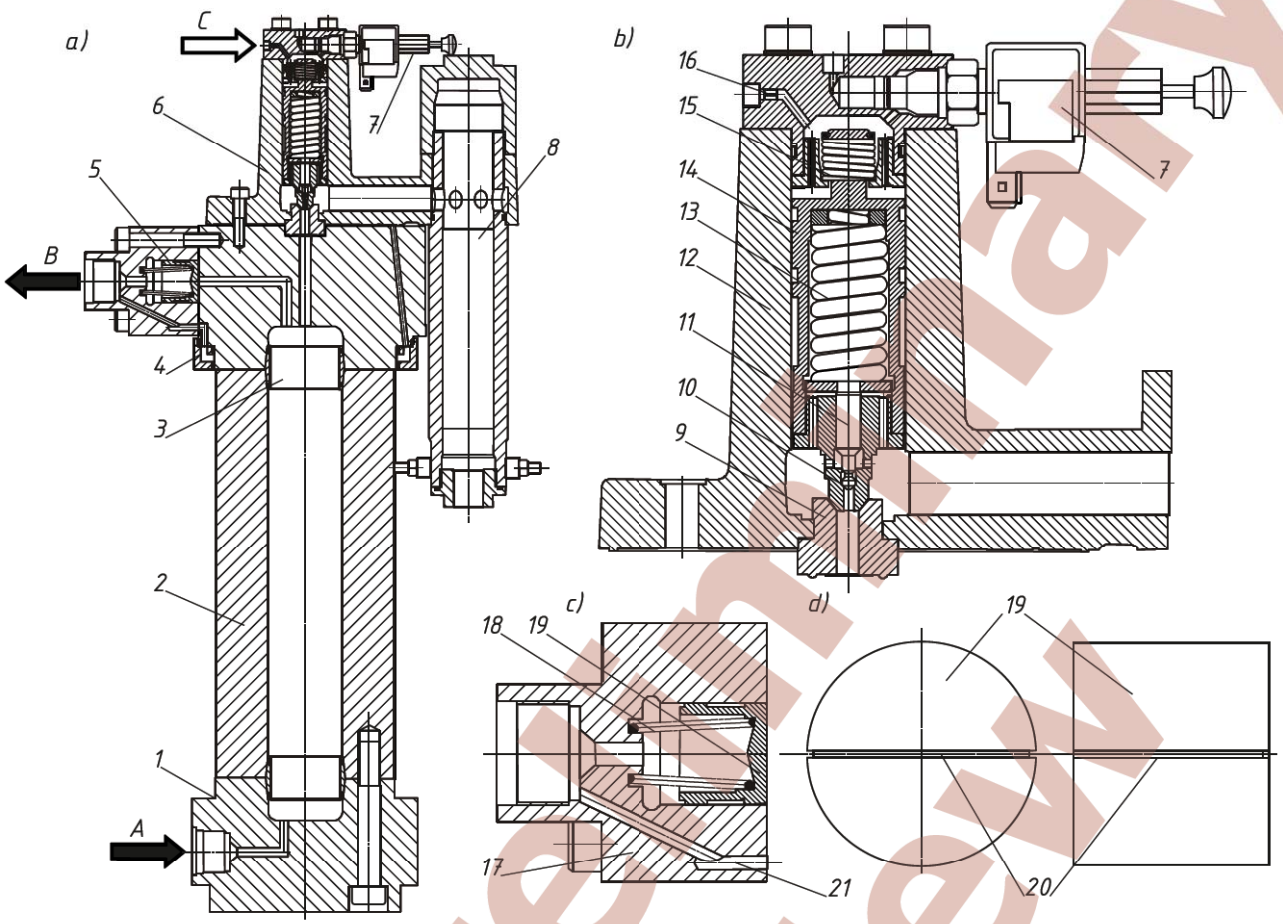


Figure 5.101 – Accumulator (*a*), pressure regulator-limiter in the accumulator (*b*), maximum cyclic fuel supply limiting valve (*c*, *d*) of Wärtsilä 46 series medium-speed engines: 1 – bottom accumulator cover; 2 – accumulator housing; 3 – seal; 4 – top cover with a leakage collection channel; 5 – valve-limiter of maximum cyclic fuel supply; 6 – pressure regulator-limiter in the accumulator; 7 – manual pressure regulator in the accumulator; 8 – buffer cavity; 9 – pressure regulator valve seat; 10 – shut-off cone of the pressure regulator valve; 11 – shut-off cone of the maximum pressure limiting valve; 12 – housing of the pressure limiter regulator; 13 – loading spring; 14 – hydraulic piston; 15 – return spring; 16 – control oil throttle; 17 – housing of the maximum cycle flow limiter valve; 18 – return spring; 19 – piston-limiter of maximum cyclic supply; 20 – throttle channels; 21 – leakage collection channel. *A* – fuel supply; *B* – fuel supply to the injectors; *C* – control oil supply (adapted from [65])

Oil into the regulator -limiter comes from the control line through a throttle insert of constant cross-section, and is discharged from the control cavity through an electrically controlled throttle valve (Fig. 5.101 *b*). Depending on the flow area of the control valve, a certain pressure is established in the cavity, which acts on the hydraulic piston. The latter, through a spring, loads the shut-off cone of the valve block separating the accumulator cavity from the low-pressure cavity. The valve block consists of a regulator valve and an emergency pressure relief valve built into its housing.

Since the movement of the hydraulic piston changes the tension of the loading spring, the opening pressure of the regulator valve changes accordingly. When the pressure exceeds the set value by approximately 15%, the valve opens and releases part of the fuel into the buffer cavity. Since the valve's closing surfaces experience significant contact loads, its seat is made of tungsten carbide. The shut-off cone of the valve is covered with the same material.

The maximum pressure limiting valve is installed inside the control valve housing. It is triggered in the event of an emergency increase in pressure in the system.

For preliminary review

When bleeding the system while parked or before starting the engine, the pressure from the hydraulic cavity of the regulator-limiter is released, the valve opens under the action of the return spring and fuel circulates freely throughout the fuel system. Relieving oil pressure from the control cavity of the regulator-limiter can be done either from the engine control panel or manually. For this purpose, there is a corresponding handle on the throttle valve.

When pressure waves exceeding 10 MPa occur in the system, the limiting valve opens and releases part of the fuel into the buffer volume (Fig. 5.101 *a*), where the wave dissipates. This ensures the stability of the fuel supply from cycle to cycle. Operating experience has shown that pressure fluctuations in the fuel system at steady state conditions do not exceed 4.0 MPa, which corresponds to 2.7...4.5% of the total pressure.

The valve-limiter of the maximum cyclic flow is made in the form of a cylindrical piston (Fig. 5.101 *c*), placed in a cylinder. The end surfaces of the piston and cylinder form the valve locking surfaces. The return of the piston to its initial position is ensured by a conical spring. There is a thin slot along the piston housing that acts as a drainage channel (Fig. 5.101 *d*). In the absence of injection, fuel flows through this channel from the accumulator cavity to the valve cavity. In this case, under the action of the spring, the piston moves to its initial position.

When injection begins, the drainage channel is not able to provide the necessary fuel flow through the valve, resulting in a pressure difference between its cavities. Under the influence of this difference, the piston moves in the cylinder. If by the time the piston reaches the extreme position, the supply does not stop, for example, due to damage to the high-pressure line, the end of the piston will rest against the bottom of the valve housing and cut off the fuel supply to the damaged line.

The leak detection system consists of a network of drainage channels in the top cover of the accumulator, each of which has its own control valve.

To increase the operational safety of the fuel system, all connecting and high-pressure pressure pipelines are double-walled. If the seals are broken, leaks are collected in the space between the walls and discharged into the drainage system, which has a separate channel for each tube.

All channels are connected through a system of control valves to a leakage collection line in which a control sensor is installed. If any of the seals are damaged, the pressure in the drainage channel increases, affecting the control valve piston. The valve, overcoming the force of the ball retainer, moves down, connecting the drainage channel with the leakage collection line. A sensor installed in the line signals a malfunction, and the protrusion of the stem of the corresponding valve indicates the location of the damage (Fig. 5.102).

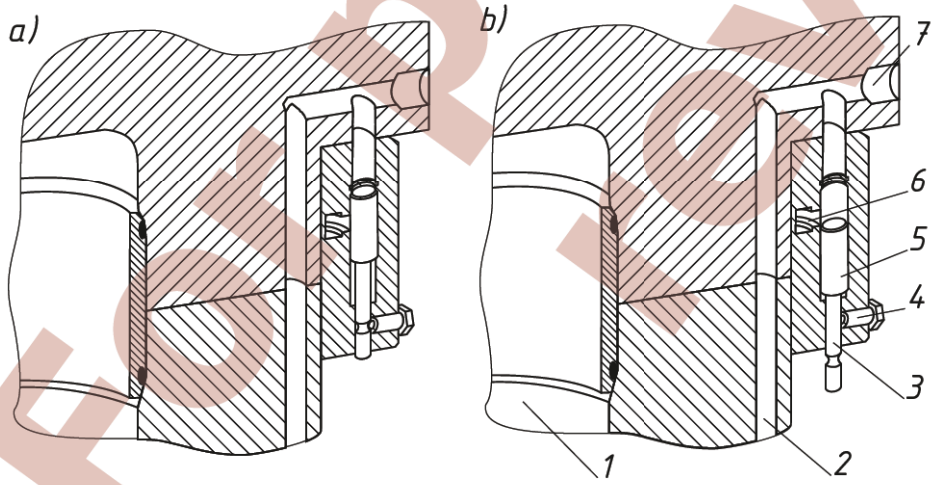


Figure 5.102 – Leak detection system: 1 – accumulator cavity; 2 – drainage channel; 3 – valve stem; 4 – spring clamp; 5 – control piston; 6 – leakage collection channel; 7 – plug; *a* – initial state; *b* – in the presence of leaks (adapted from [65])

The pumping valve is hydraulically driven from the control oil system (Fig. 5.103) and serves to ensure free circulation of fuel throughout the system during pumping and to maintain a given pressure in the buffer cavity of the batteries.

The fuel injector with electromagnetic supply control is an electro-hydraulic device, the hydraulic diagram of which is shown in Fig. 5.104.

A longitudinal section of the fuel injector in three main sections is shown in Fig. 5.105.

For preliminary review

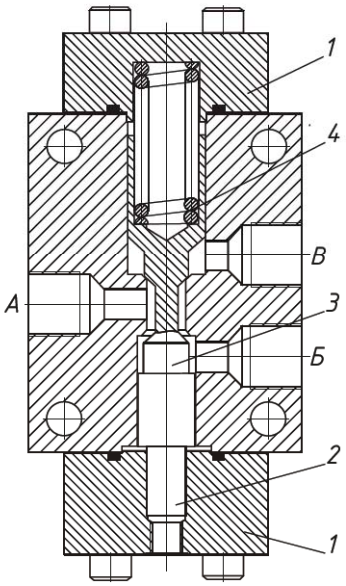


Figure 5.103 – Fuel pumping valve: 1 – cover; 2 – hydraulic drive piston; 3 – valve; 4 – return spring. *A* – oil supply from the buffer cavity; *B* – fuel discharge to the mixing tank; *C* – fuel drainage (adapted from [65])

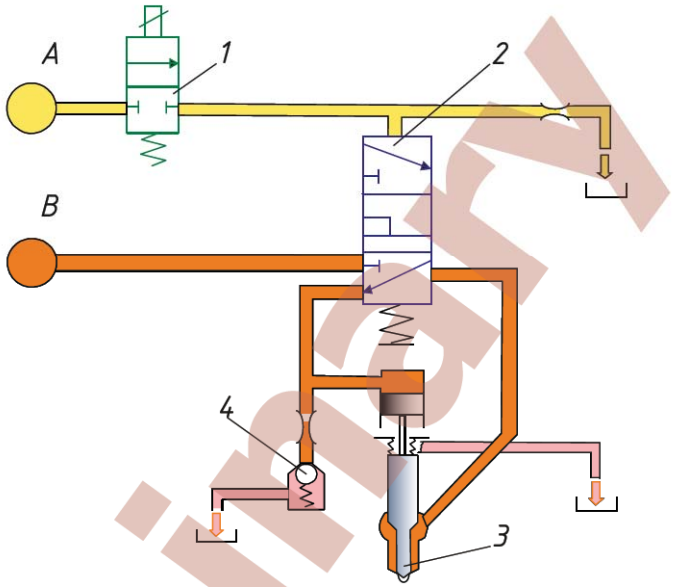


Figure 5.104 – Hydraulic diagram of the fuel injector: 1 – start valve; 2 – main valve: 3 – nozzle valve; 4 – residual pressure limiter valve; *A* – control oil; *B* – fuel (adapted from [66])

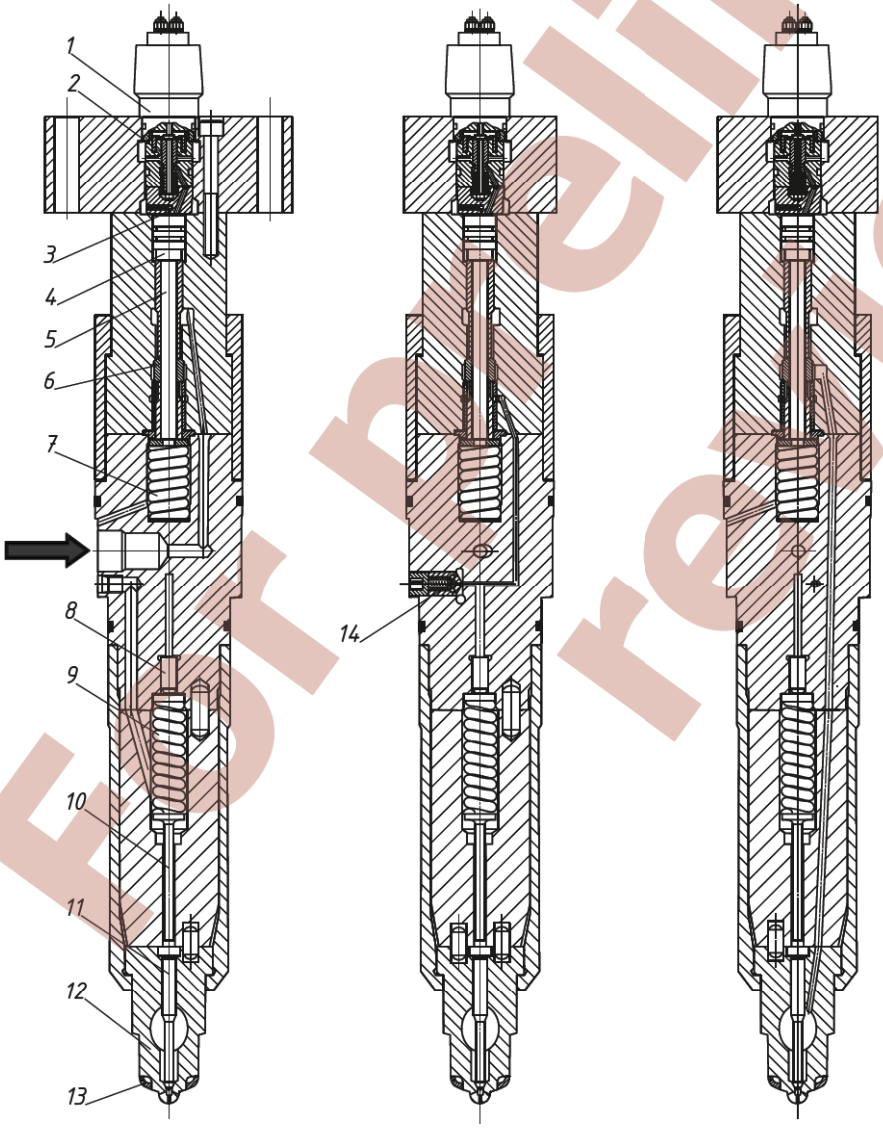


Figure 5.105 – Electrically controlled injector of medium-speed engines 46 series from Wärtsilä: 1 – solenoid; 2 – start valve; 3 – throttle hole; 4 – main valve drive piston; 5 – main valve guide; 6 – main valve; 7 – return spring; 8 – piston for hydraulic locking of the nozzle needle; 9 – loading spring of the needle valve of the nozzle; 10 – rod; 11 – needle valve of the nozzle; 12 – nozzle housing; 13 – nozzle cooling cavity; 14 – residual pressure limiter valve; *A* – fuel; *B* – oil (adapted from [65, 66])

For preliminary review

The fuel injector includes three main valves:

- starting, with electromagnetic drive;
- main, hydraulically driven from the starting valve;
- needle valve of the nozzle.

In addition, the fuel injector design includes a valve that limits the residual fuel pressure in the working cavity.

Wärtsilä has used a two-level control system in its accumulator system.

At the first level there is a start valve, which is controlled by the electronic engine control unit. This valve affects the flow of control oil supplied to it from the control line.

At the second level there is a main valve, which is hydraulically actuated by the control oil system, which is controlled by the start valve.

The design of the main valve allows it to sequentially control the hydraulic locking system of the nozzle needle and the process of fuel injection into the combustion chamber. This prevents premature fuel injection until the pressure in the fuel injector cavity reaches its maximum value. This solution has significantly improved the quality of fuel atomization, especially at the initial stage of the injection process.

The supply control start valve is electromagnetically driven by the electronic engine control unit. The working element is a cone-type locking device, held closed by a loading spring. A general view of the starting and main valve is shown in Fig. 5.106.

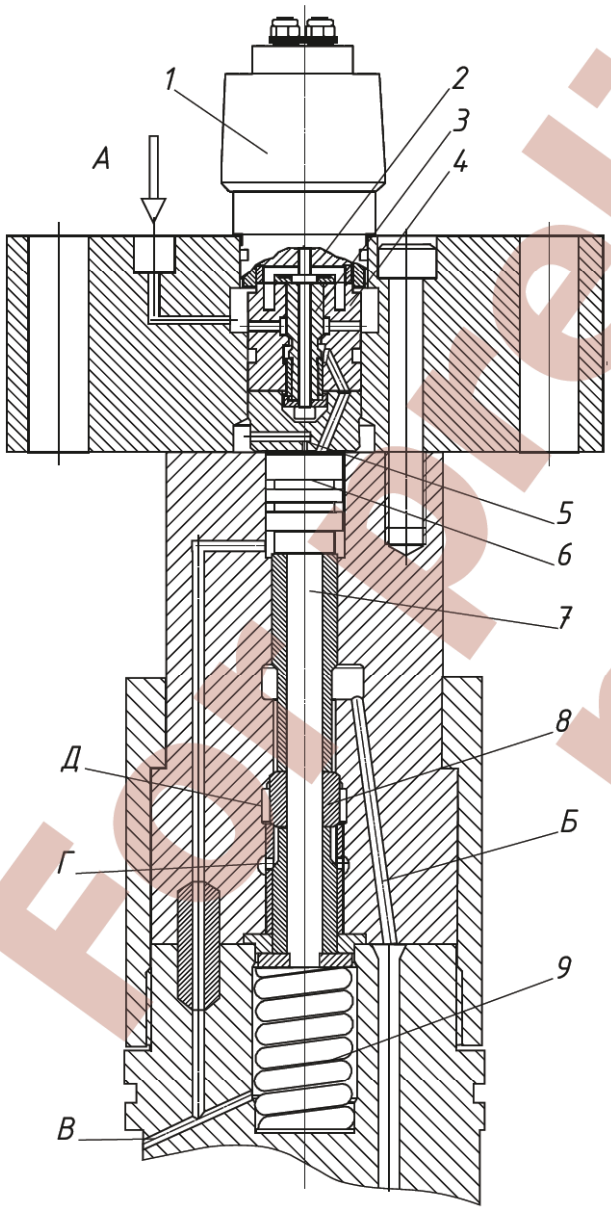


Figure 5.106 – General view of the starting and main valve: 1 – solenoid; 2 – solenoid armature; 3 – start valve; 4 – starting valve housing; 5 – throttle hole; 6 – main valve drive piston; 7 – main valve guide; 8 – main valve; 9 – return spring. *A* – control oil; *B* – fuel supply; *C* – collection of leaks; *D* – fuel outlet for hydraulic locking of the needle; *E* – fuel outlet to the nozzle (adapted from [65, 66])

For preliminary review

When a control signal is applied to the solenoid coil, its armature, overcoming the force of the loading spring, opens the valve. The cavity of the control oil line is connected to the above-piston space of the main valve hydraulic drive cylinder. The oil entering the cylinder cavity acts on the piston, moving it along with the main valve guide.

To drain oil from the cavity of the hydraulic drive cylinder, there is a throttle hole in its bottom. When the valve opens, the oil supply significantly exceeds its outflow through the throttle, and when closed, the oil gradually flows into the drain line and the piston returns to its original position.

The main valve sequentially controls the hydraulic locking of the fuel injector needle and fuel injection.

The general layout of the main valve is shown in Fig. 1.106.

A special insert made of tungsten carbide is used as a working element, which is placed on the central rod along with spacer bushings. Together with the stem, the spacer bushings form the main valve guide.

The conical seats of the main valve are also coated with tungsten carbide, one of which is located in the housing, and the other in the insert sleeve.

The need to use materials with increased hardness is caused by the fact that, to ensure the required speed, the valve has a very small stroke, and at the moment of opening and closing the gap between the seat and the valve is commensurate with the size of microparticles that are somehow present in the fuel. At this moment, particles carried away by the fuel flow interact with the working surfaces of the valve, causing their increased wear.

The use of super-hard materials and coatings allowed the company to provide a service life of the main valve of up to 30 thousand hours.

The operating procedure of the main valve is shown schematically in Fig. 5.107.

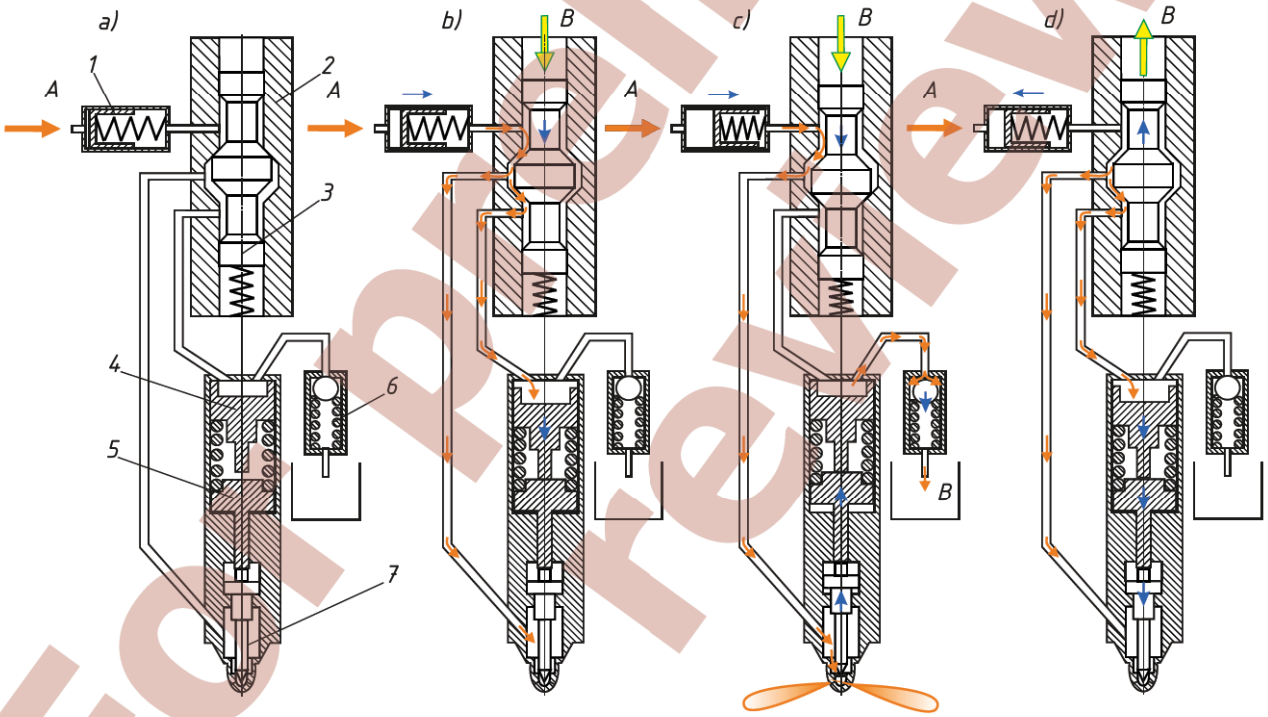


Figure 5.107 – Diagram of the main valve operation: *a* – no injection; *b* – preparation for injection; *c* – injection; *d* – supply cut-off; 1 – maximum cyclic supply limiter; 2 – main valve housing; 3 – main valve; 4 – hydraulic piston with intermediate stop of the fuel injector needle; 5 – hydraulic piston with the main stop of the fuel injector needle; 6 – residual pressure limiter valve; 7 – needle valve of the nozzle. *A* – fuel supply; *B* – control oil; *C* – fuel to drain (adapted from [65, 66])

In the absence of a control signal, there is no pressure in the piston cavity of the hydraulic drive of the main valve and, under the action of the spring, the valve occupies the uppermost position. The upper working edge of the valve blocks access of fuel to the nozzle and the hydraulic locking

For preliminary review

piston of the fuel injector needle (Fig. 5.107 *a*).

When the start valve opens, control oil enters the piston cavity of the main valve hydraulic drive. Opening slightly, the valve allows fuel to pass into the intermediate cavity, from where it enters the fuel injector cavity and into the hydraulic cylinder, the piston of which is connected to the intermediate stop of the fuel injector needle (Fig. 5.107 *b*).

The piston, overcoming the force of the return spring, lowers and through the stops blocks the opening of the needle valve of the nozzle. This ensures the injection delay necessary to achieve maximum pressure in the fuel injector cavity.

Further movement of the main valve leads to the fact that with its reverse edge it blocks the access of fuel to the hydraulic locking cylinder of the needle valve. The fuel in this cylinder is drained through the throttle hole and the residual pressure limiter valve, releasing the injector needle stop. Under the influence of pressure in the fuel injector cavity, the needle rises and fuel injection begins (Fig. 5.107 *c*).

After the start valve is closed, the control oil is drained from the main valve drive hydraulic cylinder. This causes the main valve to open slightly and reconnect the intermediate cavity with the hydraulic lock cylinder of the nozzle needle valve. Under the influence of fuel pressure, the piston moves and closes the needle valve of the nozzle through an intermediate stop, providing a sharp cut-off of the supply (Fig. 5.107 *c*). The main valve then closes, separating the injector cavity from the accumulator cavity, and the remaining fuel is drained from the injector cavity through the residual pressure limiter valve.

The electronic engine control system is a complex that includes several microprocessor modules designed to control the engine and monitor its operating parameters. The system also includes various types of sensors, with the help of which information necessary for decision-making is collected, and actuators, through which the engine is controlled. This system was codenamed WECS 7500. As an independent system, it includes a control and information display system, called WECS 2000. The general arrangement of elements of the control and monitoring system on the engine is shown in Fig. 5.108.

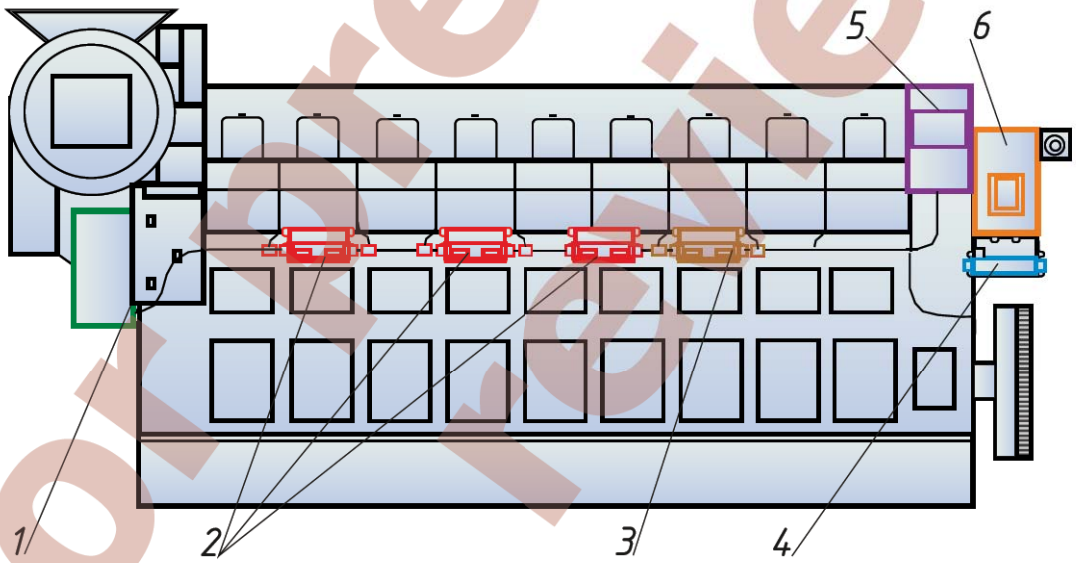


Figure 5.108 – Location of the main elements of the WECS 7500 engine control and monitoring system: 1 – power supply; 2 – controllers for controlling working cylinders; 3 – fuel system control controller; 4 – main controller; 5 – control system WECS 2000; 6 – information exchange and display module (adapted from [67, 68])

All elements of the system are redundant and redundant, so the failure of one module does not lead to engine shutdown, and the functions of the damaged module are distributed among the remaining elements of the system.

The same applies to information collection sensors, some of which have multiple redundancies. First of all, this concerns crankshaft position sensors, without which engine operation is in principle

For preliminary review

impossible. One engine is equipped with two sensors for the current position of the crankshaft, the main one and the backup one, and two sensors that mark TDC and BDC. The diagram for connecting sensors to the engine control and monitoring system is shown in Fig. 5.109.

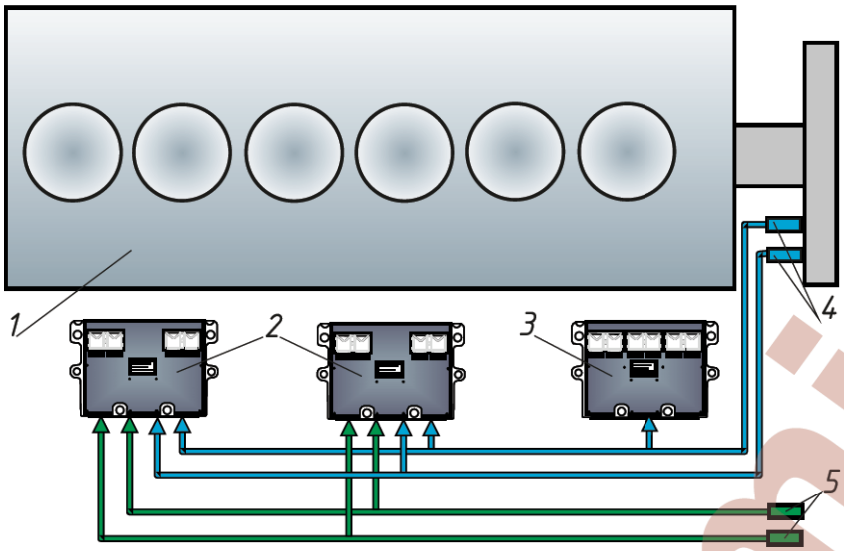


Figure 5.109 – Scheme for connecting sensors to the engine control and monitoring system: 1 – engine; 2 – controllers for controlling working cylinders; 3 – fuel system control controller; 4 – crankshaft position sensors; 5 – crankshaft rotation phase sensors (adapted from [67, 68])

The first two sensors take a signal from a special disk mounted on the flywheel, and the second sensors take a signal from a disk mounted on the end of the camshaft.

To generate a signal, 119 indentations are made on the flywheel disk every 3° and one indentation is skipped to form a reference point, which corresponds to TDC. The second disk has a stepped protrusion, the edges of which correspond to the position of the engine shaft at TDC and BDC (Fig. 5.110).

Since the camshaft rotates twice as slow as the crankshaft, the signals from this sensor coincide with the beginning of each new stroke.

The WECS system uses proprietary software specifically designed to control engines with a accumulator fuel supply system.

An original solution to the problem of preventing violations in the operation of software is to link all processes of both monitoring and control in the engine to the rotation of the so-called «virtual flywheel».

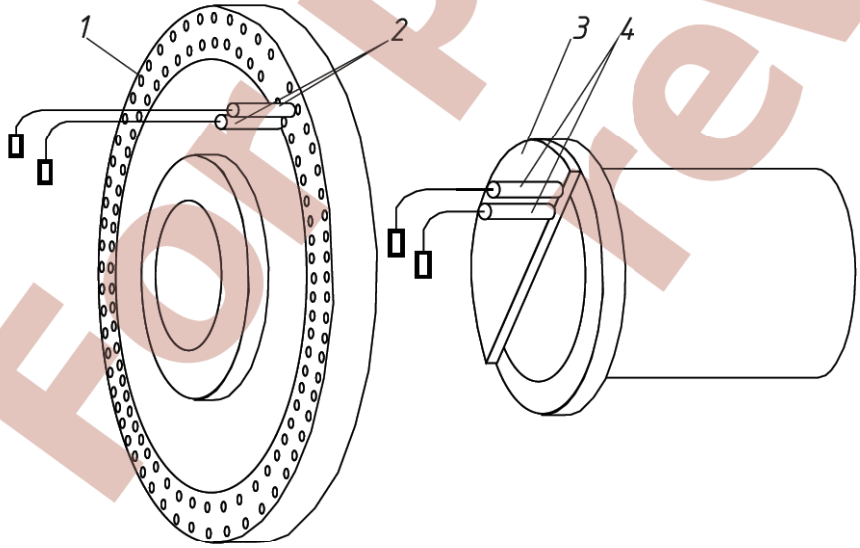


Figure 5.110 – Installation diagram of crankshaft position sensors: 1 – current position disk; 2 – current position sensors; 3 – phase disk; 4 – phase sensor (adapted from [67, 68])

A serious problem when developing software for engine control is the instability of signals from crankshaft position sensors. This is caused, firstly, by the cyclical nature of the processes, and secondly, by torsional vibrations in the «crankshaft – shaft line» system. In different modes, signals

For preliminary review

from the sensors may not correspond to the actual position of the shaft. In conditions of such instability, it is difficult to link any control action to the real process. For example, a decrease in the time between passing two nearby control points on the flywheel can be perceived by the control system as the beginning of engine acceleration. The system should respond to this by reducing the fuel supply to the cylinder. But if the decrease in time is caused not by acceleration of the shaft, but by its twisting, a decrease in supply will lead to a decrease in rotation speed, to which the system must respond by increasing supply. As a result, the so-called «engine swing» may occur, in which its speed begins to change cyclically with increasing amplitude. If no measures are taken, this could result in a serious accident.

In the WECS control system, all processes are synchronized relative to a virtual flywheel, which, depending on the operating mode, «rotates» at a constant speed or with constant acceleration. The virtual one is synchronized with the real flywheel only through position and phase sensors. Moreover, synchronization is maintained even if three out of four sensors fail.

Integrated control of the engine's operating process is carried out by changing the supply law in two main positions:

- regulation of the beginning and duration of fuel supply to the working cylinder;
- change in fuel pressure in the accumulator and in front of the nozzle.

The algorithm for changing the above parameters, implemented in Wärtsilä engines, is shown in Fig. 5.111.

Changing the fuel supply advance angle makes it possible to optimize the heat release process relative to TDC, ensuring constancy of the maximum cycle pressure in the load range of 75...100% of the nominal value.

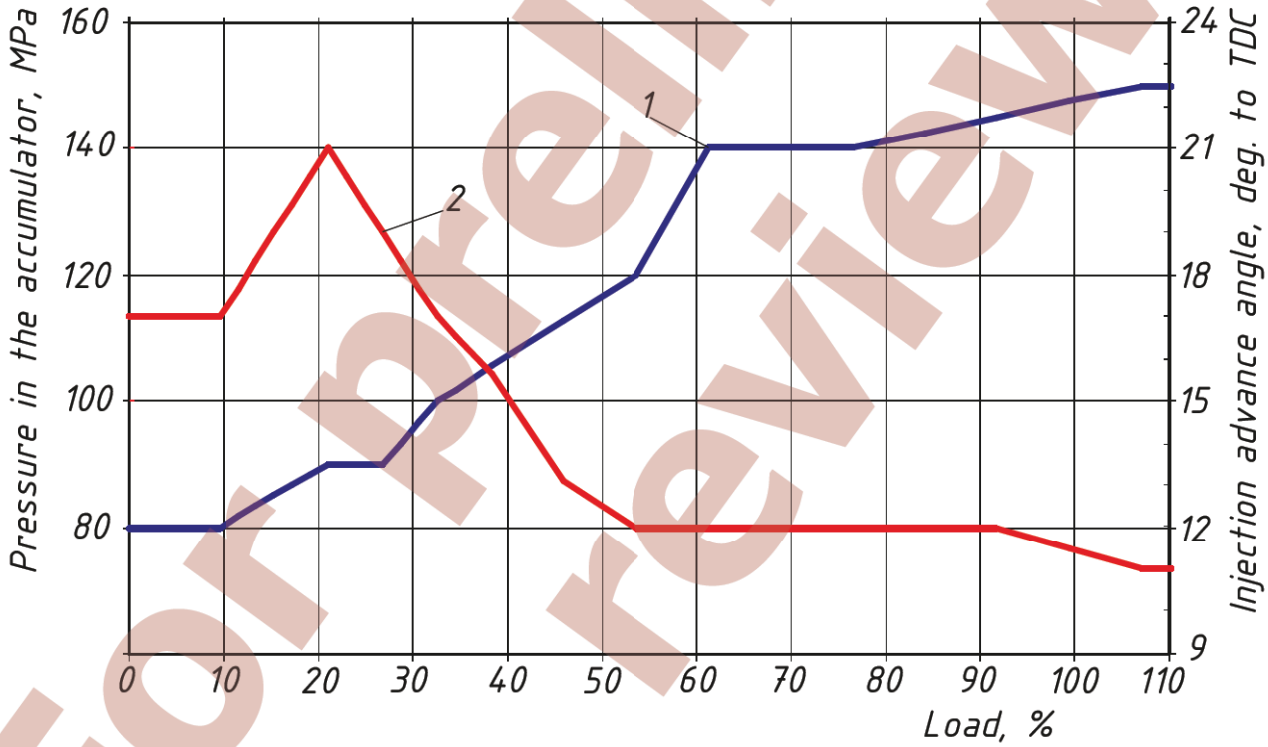


Figure 5.111 – Change in the supply advance angle and pressure in the accumulator as a function of engine load: 1 – pressure in the accumulator; 2 – injection advance angle (adapted from [68])

Regulating the pressure in the accumulator allows you to control the quality of fuel atomization. At low cyclic supplys, a decrease in injection pressure makes it possible to increase the depth of penetration of fuel jets, which improves the distribution of fuel throughout the volume of the combustion chamber. An increase in load leads to the fact that cyclic supplys increase. At constant pressure, this can lead to the settling of a significant part of the fuel on the walls of the combustion chamber.

For preliminary review

An increase in injection pressure leads to an increase in atomization fineness, as a result of which the penetration range of fuel jets decreases.

Currently, the accumulator injection system is installed on Wärtsilä diesel engines of the 32, 38, 46 and 46F.

5.7.3.3 The Caterpillar accumulator fuel supply system is installed on medium-speed MaK engines. The system was based on technical solutions that the company had previously worked on its high-speed engines. When designing the system, the goal was to retain all the advantages of accumulator systems, simplify its design as much as possible, minimizing the number of connections of elements operating under high pressure. A general view of the accumulator power supply system for MaK engines is shown in Fig. 1.112.

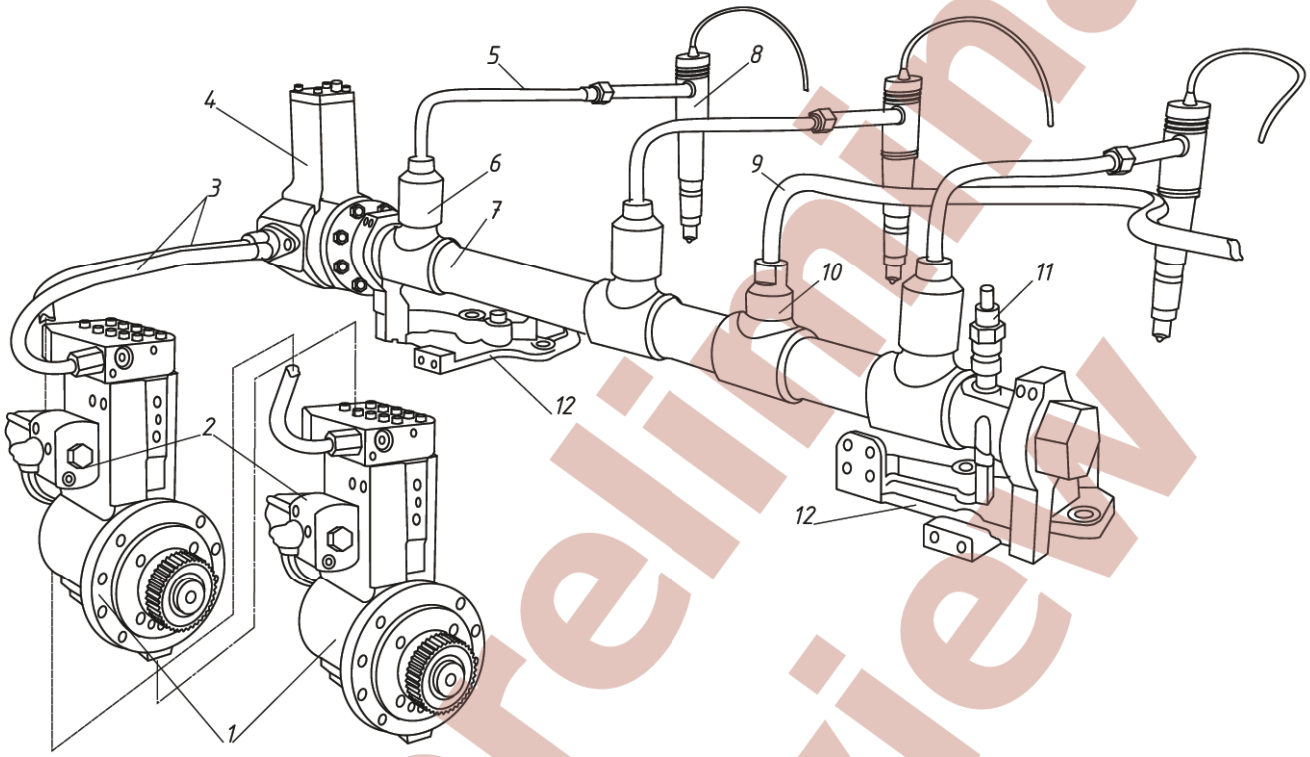


Figure 1.112 – General view of the accumulator power supply system for MaK engines: 1 – high pressure fuel pumps; 2 – electromagnetic valves for supply control; 3 – fuel supply pipeline from the pumps to the accumulator; 4 – pressure limiter regulator; 5 – pipeline for supplying fuel to the injectors; 6 – union nut for fastening the pipeline; 7 – accumulator; 8 – electrically controlled fuel injector; 9 – connecting pipeline between the accumulator sections; 10 – fastening of the connecting pipeline; 11 – pressure sensor; 12 – mounting bracket (adapted from [69])

During the design, the system was adapted to operate on both distylates and heavy residual fuels.

Caterpillar accumulator fuel injection system from the previously discussed systems is that manufacturers have abandoned the use of modernized pump sections used in traditional injection systems as high-pressure pumps. To supply fuel under high pressure, separate double-plunger pumps are used here, which are installed on a special drive box and driven from the camshaft drive gears. To provide the required performance, each pump has two four-jaw washers. In addition, the pump speed is higher than traditional camshaft driven designs. As a result of this, one pump is able to provide the necessary fuel supply to the accumulator in all engine operating modes, however, for greater reliability, two pumps of the same type are installed on the engine. The pumps are driven through a gear train equipped with a safety friction clutch, which, in the event of jamming, is destroyed, ensuring that the damaged pump is turned off.

High pressure pumps used by Caterpillar have a valve control system with solenoid valve actuators. The pump supply is controlled by an electronic engine control unit. The pump diagram is shown in Fig. 5.113, and the order of its operation is shown in Fig. 5.115.

For preliminary review

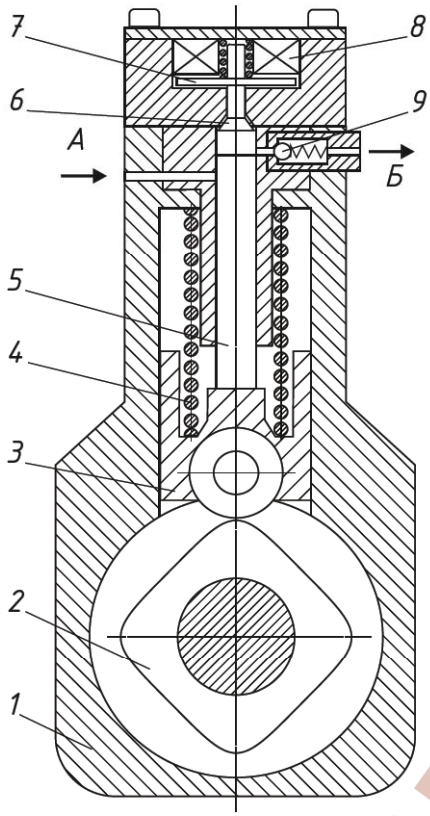


Figure 5.113 – Diagram of a pump with electromagnetic supply control: 1 – pump housing; 2 – drive cam; 3 – roller pusher; 4 – return spring; 5 – plunger; 6 – supply control valve; 7 – solenoid armature; 8 – solenoid; 9 – delivery valve; 10 – plunger bushing. *A* – fuel supply; *B* – fuel supply to the accumulator (adapted from [9])

If there is no signal from the control unit, the control valve is in the open state under the action of the return spring. When the plunger moves upward, fuel from the cavity above the plunger flows back into the filling line through the open valve – the plunger idles. The active pumping process begins when a control signal is applied to the solenoid, the valve closes and is held in this position by forces arising from fuel pressure. Through the delivery valve, the fuel remaining in the above-plunger cavity enters the accumulator.

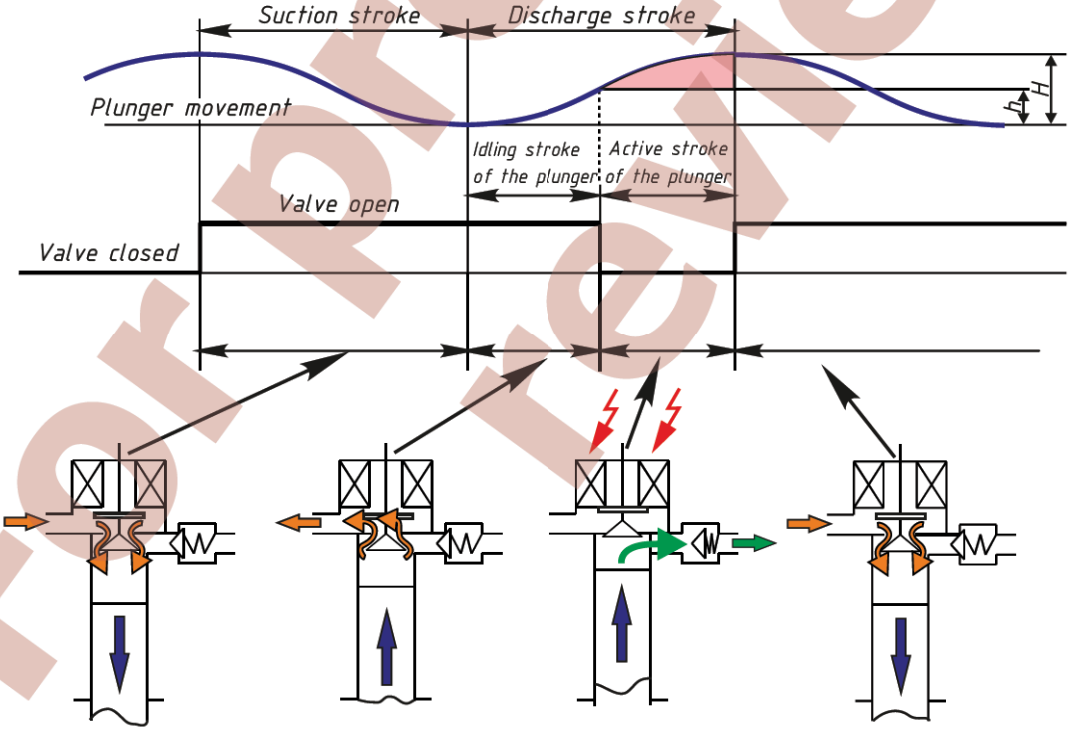


Figure 5.114 – Operating order of an in-line plunger pump with valve control of flow rate (adapted from [70])

A significant advantage of such regulation is more precise control of pump flow, resulting in reduced costs for their mechanical drive. In addition, the costs of electric valve drives are reduced,

For preliminary review

since a short current pulse is required to close them, and then the valve is kept closed automatically by pressure in the cavity above the plunger.

The pressure accumulators used by the company are a thick-walled pipe placed in a protective casing. The space between the accumulator and the protective casing is used to collect leakage from the connections.

To prevent the occurrence of pressure waves of significant amplitude in the accumulator, it is made in the form of separate sections connected to each other by double-walled high-pressure pipelines. One accumulator section supplies fuel to three or four engine cylinders, depending on the total number of cylinders.

To protect the inner tube of the accumulator from stresses caused by the presence of radial holes, special bandages are used into which the fuel line fittings are screwed. The bands absorb radial forces, and the holes themselves with this design have the minimum required diameter to ensure the specified throughput (Fig. 5.115).

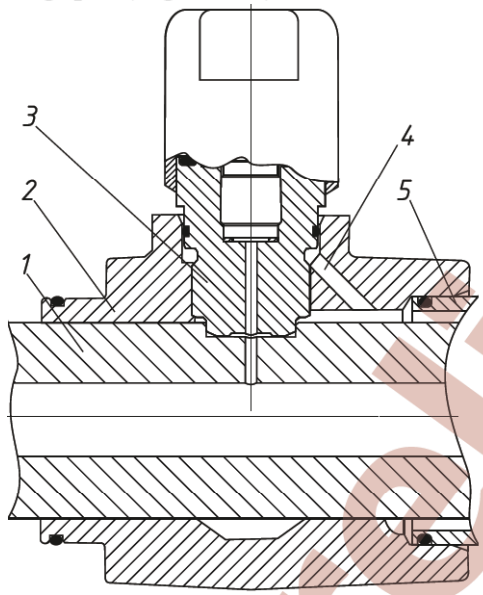


Figure 5.115 – Fragment of a pressure accumulator with an external bandage for fastening high-pressure pipelines: 1 – accumulator housing; 2 – bandage; 3 – fitting; 4 – drainage channel for collecting leaks; 5 – external accumulator casing (adapted from [9])

Injectors. An electrically controlled injector with an electromagnetic drive is used for fuel injection. To simplify the design, the company abandoned the drive of the main spool using control oil, using for these purposes directly the fuel supplied to the injector.

To protect the solenoid valves from high temperatures, an oil injector cooling system was used, which, however, is traditional for MaK engines. For engines equipped with a common rail system, the oil cooling of the injectors is more developed and extends not only to the tip of the fuel injector, but also to the electrical part.

5.7.4 Accumulator injection systems for high-speed marine diesel engines

Accumulator injection systems for high-speed marine diesel engines are quite diverse both in circuit solutions and in the design of individual elements. Therefore, the most common design options for these systems will be considered below. The configuration of the fuel system is highly dependent on the cylinder power of the engine.

The main elements of HSE accumulator injection systems are:

- fuel pump module, which includes low and high pressure pump sections;
- electronic engine control unit with a sensor system to monitor its operating parameters;
- pressure accumulator with connecting fittings;
- high pressure pipelines;
- electrically controlled fuel injection fuel injectors.

In addition, the system includes elements for fuel purification.

The diagram of the accumulator-type fuel system of a small-sized high-speed diesel engine is shown in Fig. 5.116, the general arrangement of individual elements on the engine is shown in Fig. 5.117.

For preliminary review

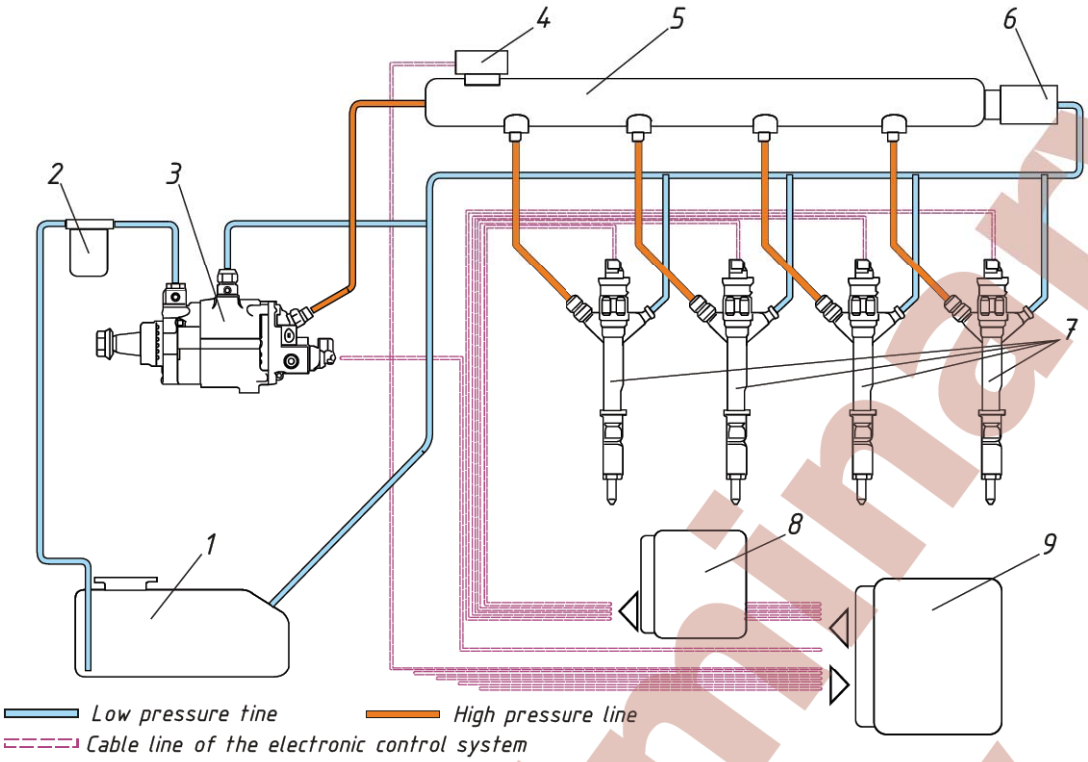


Figure 5.116 – General diagram of the accumulator fuel injection system of a high-speed diesel engine: 1 – fuel tank; 2 – fuel filter; 3 – high pressure fuel pump; 4 – fuel pressure sensor in the accumulator; 5 – accumulator; 6 – fuel pressure regulator in the accumulator; 7 – injectors with electromagnetic control; 8 – injector control unit; 9 – engine control unit (adapted from [71])

To fuel Filters have requirements similar to those of traditional fuel systems.

If the engine is supposed to use not only distillate fuels, the system is equipped with fuel heaters and viscosity regulators.

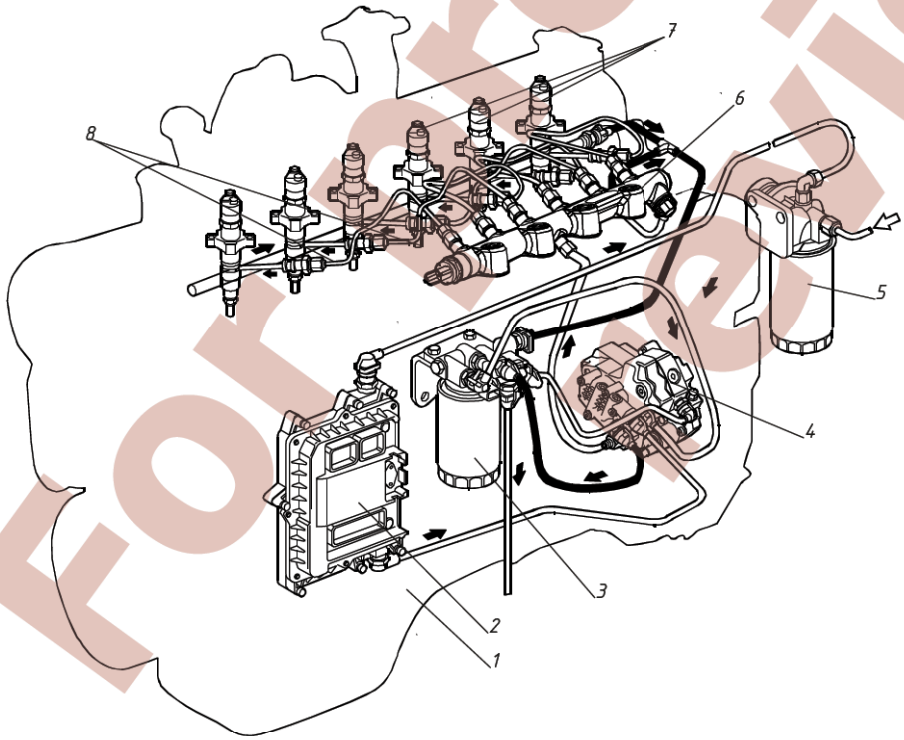


Figure 5.117 – General arrangement of elements of the accumulator fuel injection system of a high-speed diesel engine: 1 – engine; 2 – electronic engine control unit; 3 – fine fuel filter; 4 – pump module; 5 – fuel coarse filter; 6 – accumulator; 7 – injectors with electromagnetic control; 8 – high pressure pipelines (adapted from [72])

Fuel pump block serves to supply fuel to the accumulator at a given pressure. It includes a low pressure module with booster pump and a high pressure module. Further, for simplicity, we will simply call the fuel pump unit a high-pressure fuel pump.

For preliminary review

The fuel priming pump in the low pressure module is used to ensure the required fuel supply to the high pressure module. A supply pressure of 0.5...0.8 MPa is necessary to ensure stable operation of the high-pressure sections and effective lubrication of its parts.

Different manufacturers of fuel equipment use different types of booster pumps in their designs, the main ones of which are shown in Fig. 5.118.

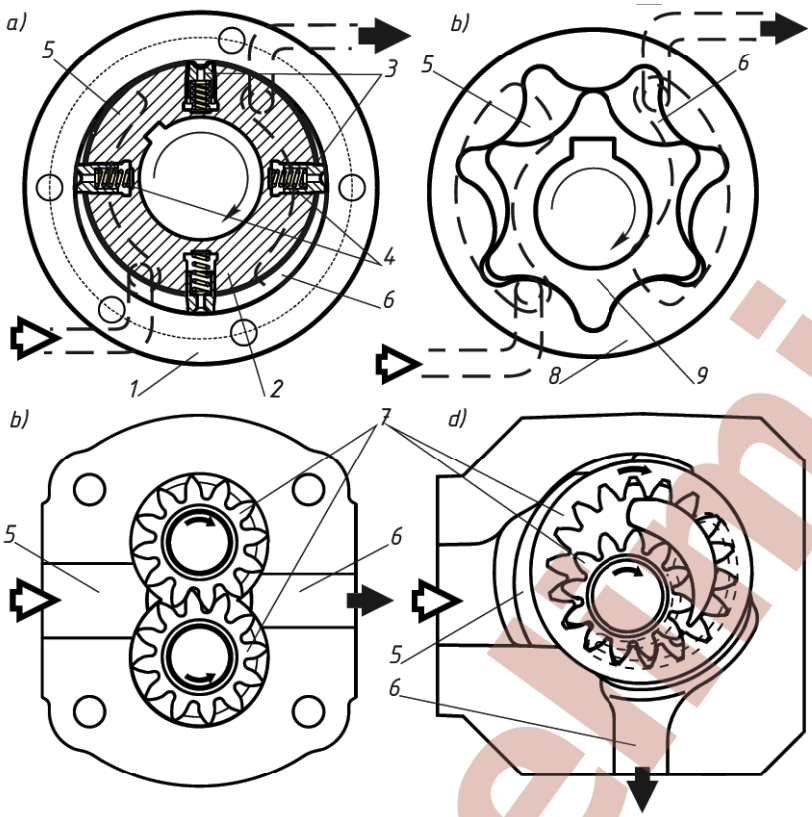


Figure 5.118 – Pumping sections of modular pumps of Common Rail systems of high-speed diesel engines: *a* – vane (vane pump); *b* – gerotor hydraulic pump; *c* – gear with external gearing *d* – gear with internal gearing; 1 – vane pump stator; 2 – rotor; 3 – working plates (gates); 4 – return springs; 5 – filling port; 6 – discharge port; 7 – working gears; 8 – rotating stator of the gerotor pump; 9 – rotor (adapted from [9, 70, 74])

Plate (gate) pumps (Fig. 5.118 *a*) operate as follows. As the shaft rotates, spring-loaded plates (called gates) inserted into slots in the rotor are pressed against the stator housing. As a result, two cavities are formed between the stator and the rotor, hermetically separated from each other by plates. As the rotor rotates, the volume of one of the cavities gradually increases (suction occurs into this cavity), the volume of the other cavity decreases proportionally (fuel is pumped from this cavity).

Vane pumps are distinguished by their simplicity and reliability in operation, as they are able to automatically compensate for the wear of the working parts – the plates. In addition, pumps of this type have self-priming capabilities.

Gerotor pumps are a type of internal gear pump. However, they have significant differences (Fig. 5.118 *b*). The stator and rotor teeth have a special trochoidal profile. The drive gear, with external gears, has the number of teeth only one less than the outer gear with internal gearing. When the inner wheel, eccentrically fixed relative to the outer one, rotates, two isolated cavities are formed between the teeth, one of which increases its volume (suction cavity), and the other reduces it (discharge cavity). Thanks to the special engagement profile, the gerotor pump eliminates the need for a separator to prevent connection between the suction and discharge chambers of the pump.

Gear pump with external gearing is a positive displacement hydraulic machine consisting of a pumping unit, a housing and a cover. The pump pump assembly consists of drive and driven gears (Fig. 5.118 *c*). The drive gear is in constant mesh with the driven gear and causes it to rotate. When the pump gears rotate in opposite directions, the teeth in the suction cavity, disengaging, create a vacuum. Due to this, fuel enters the suction cavity, which, filling the cavities between the teeth of both gears, moves them along the cylindrical walls of the wells in the housing and is transferred from the suction cavity to the discharge cavity. In the discharge cavity, the gear teeth, when

For preliminary review

meshed, push fuel out of the depressions into the discharge pipeline. The pump parts are lubricated and cooled by the pumped fuel.

Gear pumps with internal gearing are more compact than pumps with external gearing, but due to the complexity of manufacturing they are used less frequently (Fig. 5.118 *d*). To separate the suction and discharge cavities, a crescent-shaped separating element is installed between the gears, filling the space between them. The involute profile of the teeth ensures their long-term engagement, and therefore flow and pressure pulsations are minimal. This, accordingly, reduces the noise during pump operation.

The flow rate of all pumps considered is practically proportional to the shaft speed, therefore its value is limited by the bypass valve on the discharge side. A variant of the bypass valve design used in Delphi rotary plunger pumps with a ball valve is shown in Fig. 5.119.

When the pressure in the discharge cavity reaches a given value, the ball valve, overcoming the force of the loading spring, opens and transfers part of the fuel from the discharge cavity to the suction cavity.

To remove air from the fuel system before starting, a manual fuel pump can be installed directly on the fuel priming pump or in the low pressure line.

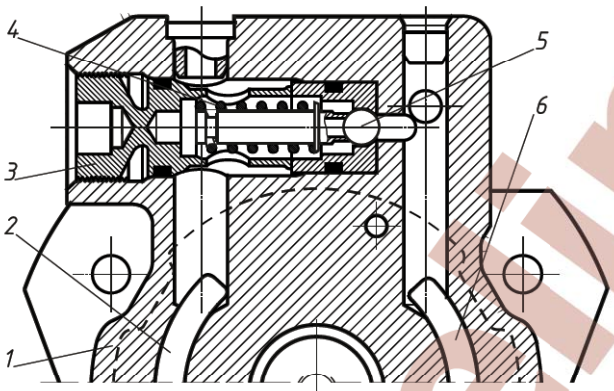


Figure 5.119 – Bypass valve of the booster pump of Common Rail high-speed diesel systems: 1 – pump housing; 2 – suction cavity; 3 – valve housing; 4 – valve loading spring; 5 – ball valve; 6 – discharge cavity (adapted from [45])

Pump modules for Common Rail systems of high-speed diesel engines have a variety of designs and layouts. In the early stages of the introduction of accumulator systems, fuel pumps were made according to a traditional block design with an in-line arrangement of high-pressure pump sections. Such pumps have a well-developed design and practical design solutions, which makes their use justified both in terms of reliability and ease of operation. An example of such a pump is the design developed by L'Orange GmbH for Common Rail systems of engines of all sizes. A general view of the pump is shown in Fig. 5.120, and its design in Fig. 5.121.

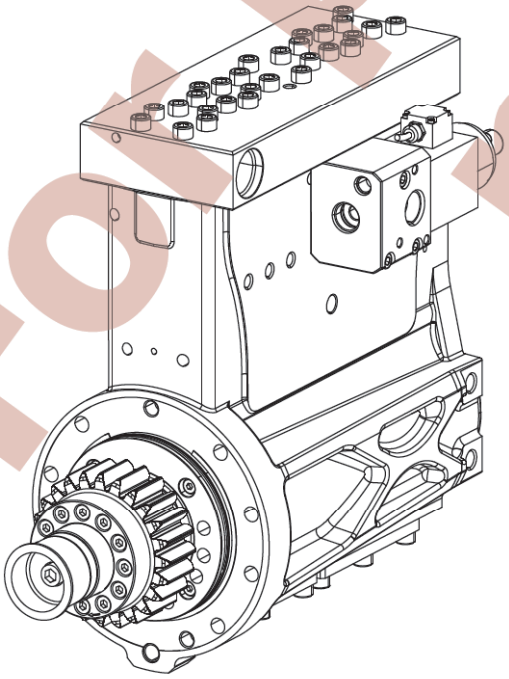


Figure 5.120 – General view of the pump from L'Orange GmbH (adapted from [57])

For preliminary review

Such pumps are used both on small two-stroke low-speed diesel engines (for example, S35ME-C9-CR from MAN X-35; X-40 from WinGD) and on powerful high-speed ones (for example, engines from MTU).

In a number of later designs, to reduce the number of plunger pairs, each of them is driven using cams having from two to four protrusions.

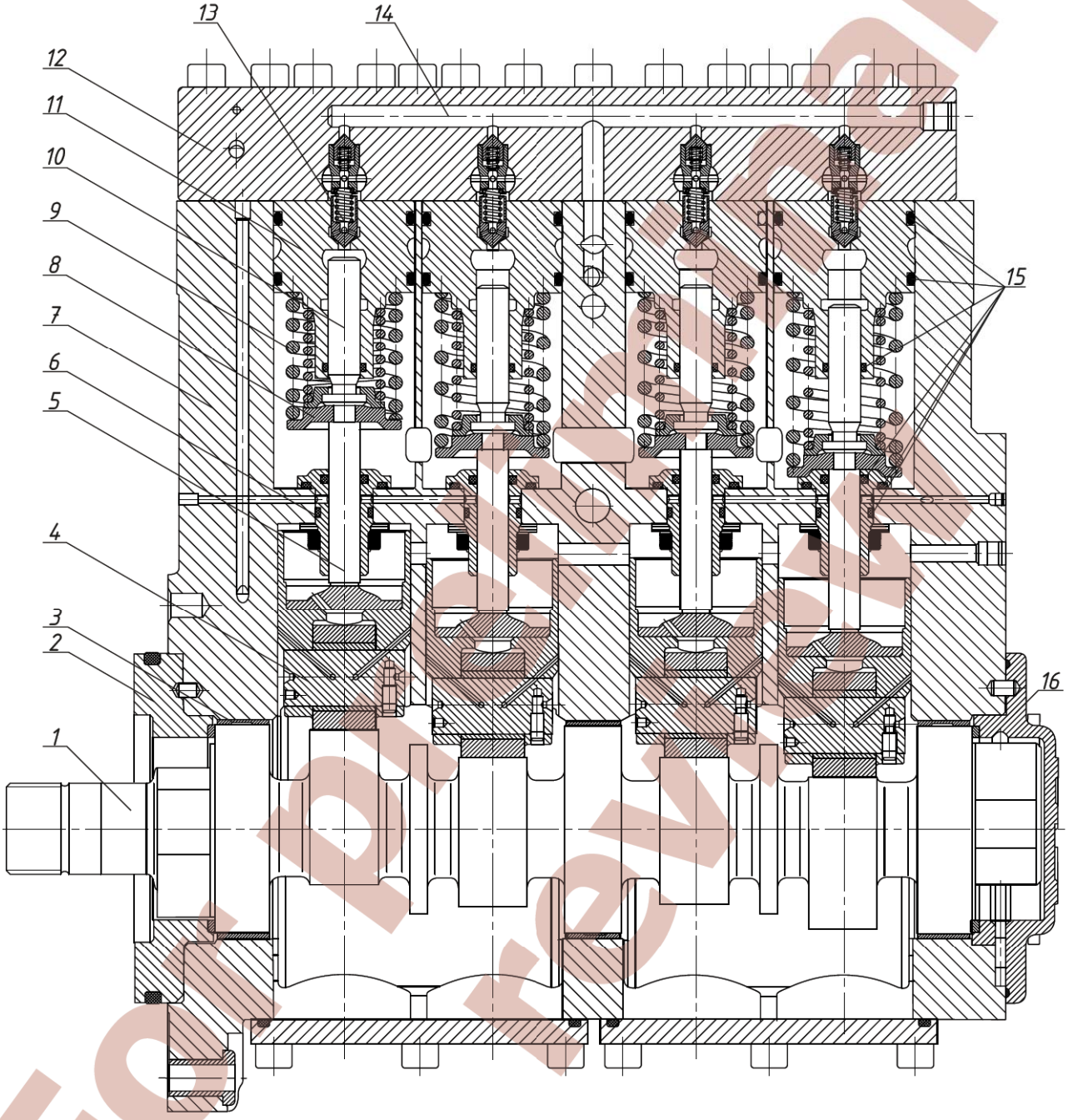


Figure 5.121 – In-line high-pressure fuel pump from L'Orange GmbH for Common Rail systems: 1 – cam shaft; 2 – front cover; 3 – support bearing; 4 – roller pusher; 5 – plunger rod; 6 – seal of the plunger rod; 7 – pump housing; 8 – lower plate of the return spring; 9 – return spring; 10 – plunger; 11 – plunger bushing; 12 – pump cover; 13 – delivery valve; 14 – high pressure line; 15 – polymer seals; 16 – back cover (adapted from [73])

In Fig. 5.122 shows a Delphi HP0 series fuel pump with an in-line arrangement of pump sections.

For preliminary review

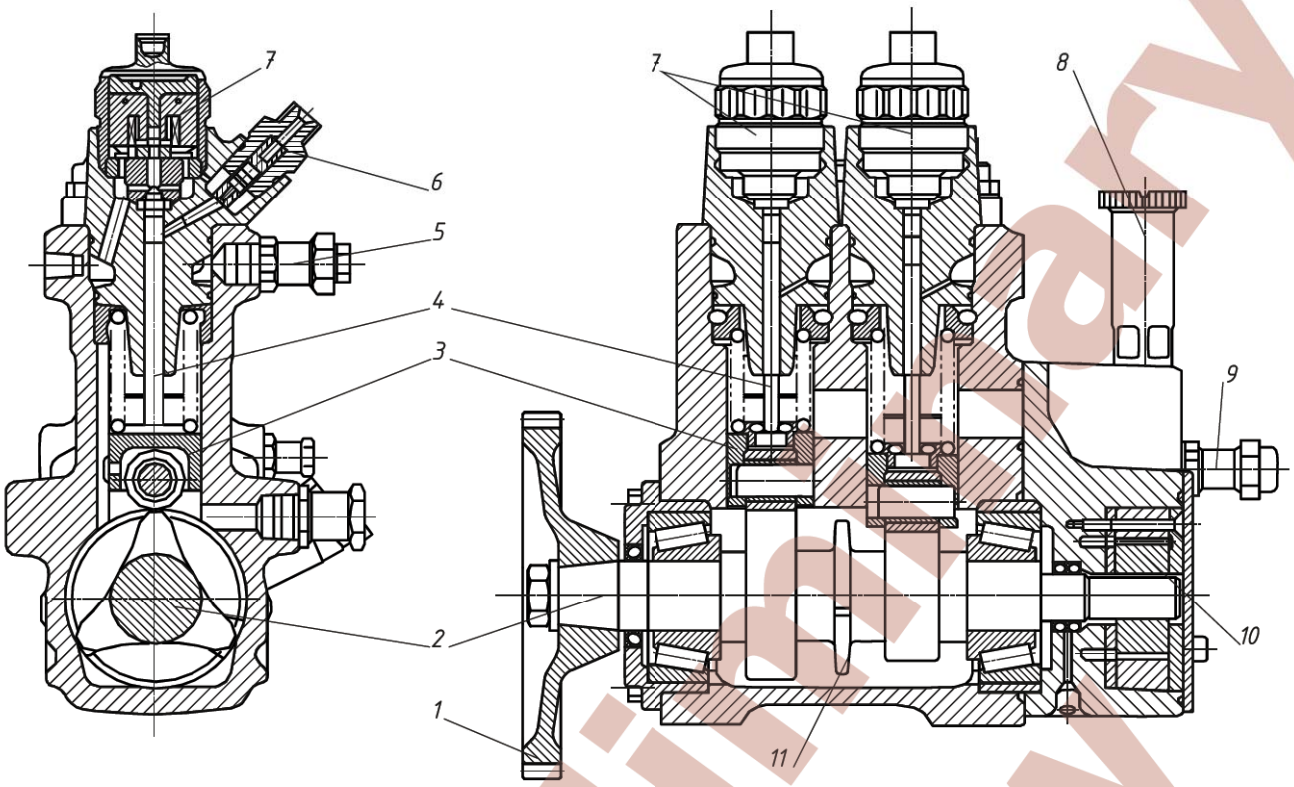


Figure 5.122 – In-line plunger pump type HP0 from Delphi for Common Rail systems of high-speed diesel engines: 1 – drive gear; 2 – drive shaft with cams; 3 – plunger pusher; 4 – plunger; 5 – bypass valve; 6 – high pressure line connection fitting; 7 – solenoid valve for supply control; 8 – booster pump; 9 – pressure limiting valve in the pumping line; 10 – booster pump; 11 – cam with a mark for the speed sensor (adapted from [70])

The supply is controlled using an electrically controlled valve according to the diagram shown in Fig. 5.114, by changing the active stroke of the plunger.

This pump is designed for power supply systems for engines with a number of cylinders from four to eight and with an operating pressure of up to 120 MPa. The pump rotation speed is two times lower than the engine speed. The required supply is ensured by installing cams with different numbers of projections (Table 5.7). Thus, the number of pump strokes coincides with the number of engine operating cycles per revolution of the crankshaft, and the moment of the injection stroke of the plunger is synchronized with the moment of fuel injection. This ensures uniformity of fuel supply and maintenance of the specified pressure in the fuel accumulator.

Table 5.7 – Characteristics of the drive sections of the Delphi type HP0 pump

Number of cylinders	Number of lugs on cam	Number of pump strokes
4	2	4
6	3	6
8	4	8

To reduce friction costs, roller pushers are used to drive the plungers.

Rotary pumps with radial plungers are carried out according to the scheme presented in Fig. 5.123.

The stator, coaxial with the drive shaft, has radial drillings into which precision-fitted plungers with roller pushers moving towards each other are inserted.

During the pumping process, translational motion is imparted to the plungers due to the pusher rollers running onto the radial protrusions of the rotor connected to the pump drive shaft. The return of the plungers to their original position is carried out by springs installed under the pusher.

For preliminary review

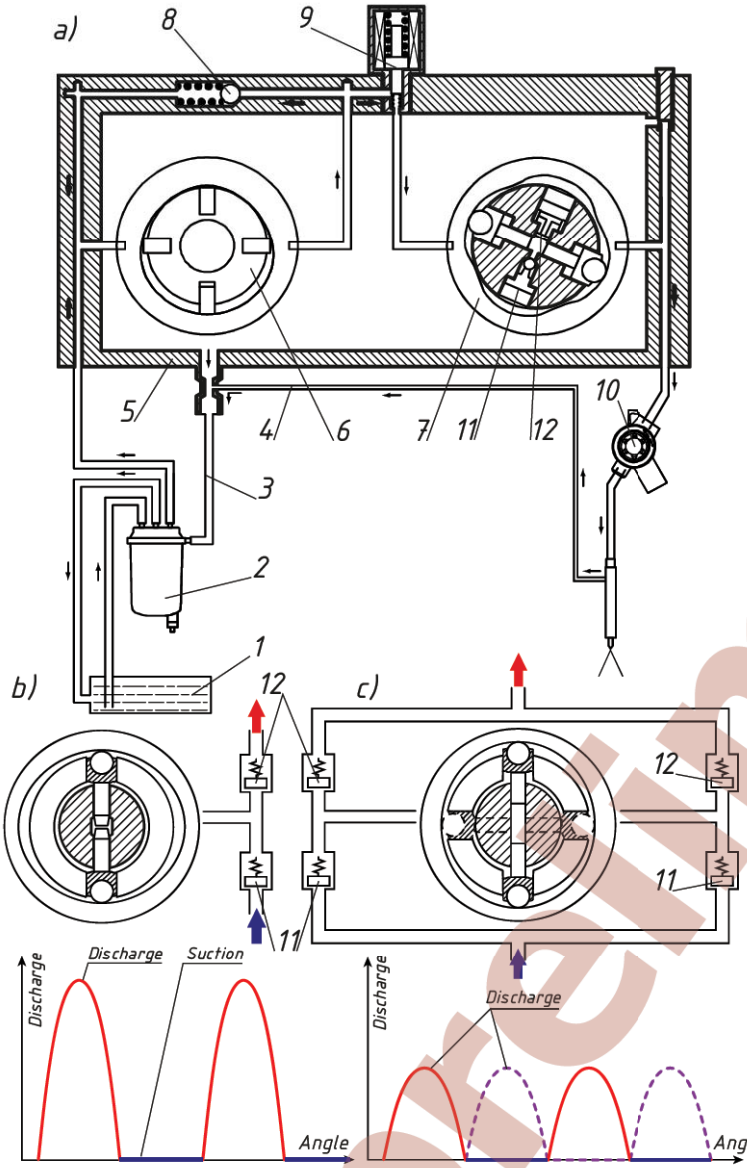


Figure 5.123 – Diagram of a rotary plunger pump (*a*) and the operating procedure of a single-section (*b*) and two-section pump (*c*) with supply characteristics: 1 – fuel tank; 2 – fuel filter; 3 – drain line; 4 – fuel drainage line from the injectors; 5 – pump housing; 6 – pump of the fuel pumping section; 7 – high pressure section; 8 – pressure limiting valve in the pumping circuit; 9 – throttling valve for adjusting the supply of the high pressure pump; 10 – accumulator; 11 – suction valve; 12 – delivery valve (adapted from [74, 75])

When the plungers diverge, the inter plunger cavity communicates through the suction valve with the low pressure line, into which fuel is supplied by a vane pump. During the discharge stroke, fuel is forced into the high pressure line through the delivery valve and then enters the accumulator.

For engines with a small cyclic flow, the pump is made with one pump section, including two plungers (Fig. 5.123 *b*). If it is necessary to ensure greater pump performance, it is equipped with two pump sections, offset from each other by 90°, each of which independently supplies fuel to the accumulator (Fig. 5.123 *c*). In this case, fuel pulsations in the high-pressure line are reduced and torque pulsations on the pump shaft are reduced, which leads to a reduction in noise and vibration during its operation.

The pump performance is changed by dosing the fuel at the inlet to the high-pressure pump module using an electrically controlled valve.

A pulse is sent from the engine control unit to the valve, opening it when the plungers of this section are on the suction stroke. The duration of the suction stroke is determined by the opening time of the control valve, which can be changed by the electronic unit depending on the engine operating mode. As a result, the degree of filling of the inter plunger space changes, and, as a result, the amount of fuel entering the accumulator changes. The disadvantage of such regulation is that when the supply decreases, the kinematic connection between the pusher roller and the generatrix of the rotor protrusions breaks. When the injection stroke begins, the kinematic connection is restored, and the process of placing the pusher roller on the rotor generatrix is accompanied by an impact.

The pumps considered have good weight and size characteristics and are capable of providing

For preliminary review

pressure in the accumulator up to 135.0 MPa.

An example of a pump of this design, the DFP1 series from Delphi, is shown in Fig. 5.124.

A general view and longitudinal section of a pump with one high-pressure section is shown in Fig. 5.124 *a* and 5.124 *b*. A fragment of a two-section high-pressure stage is shown in Fig. 5.124 *c*.

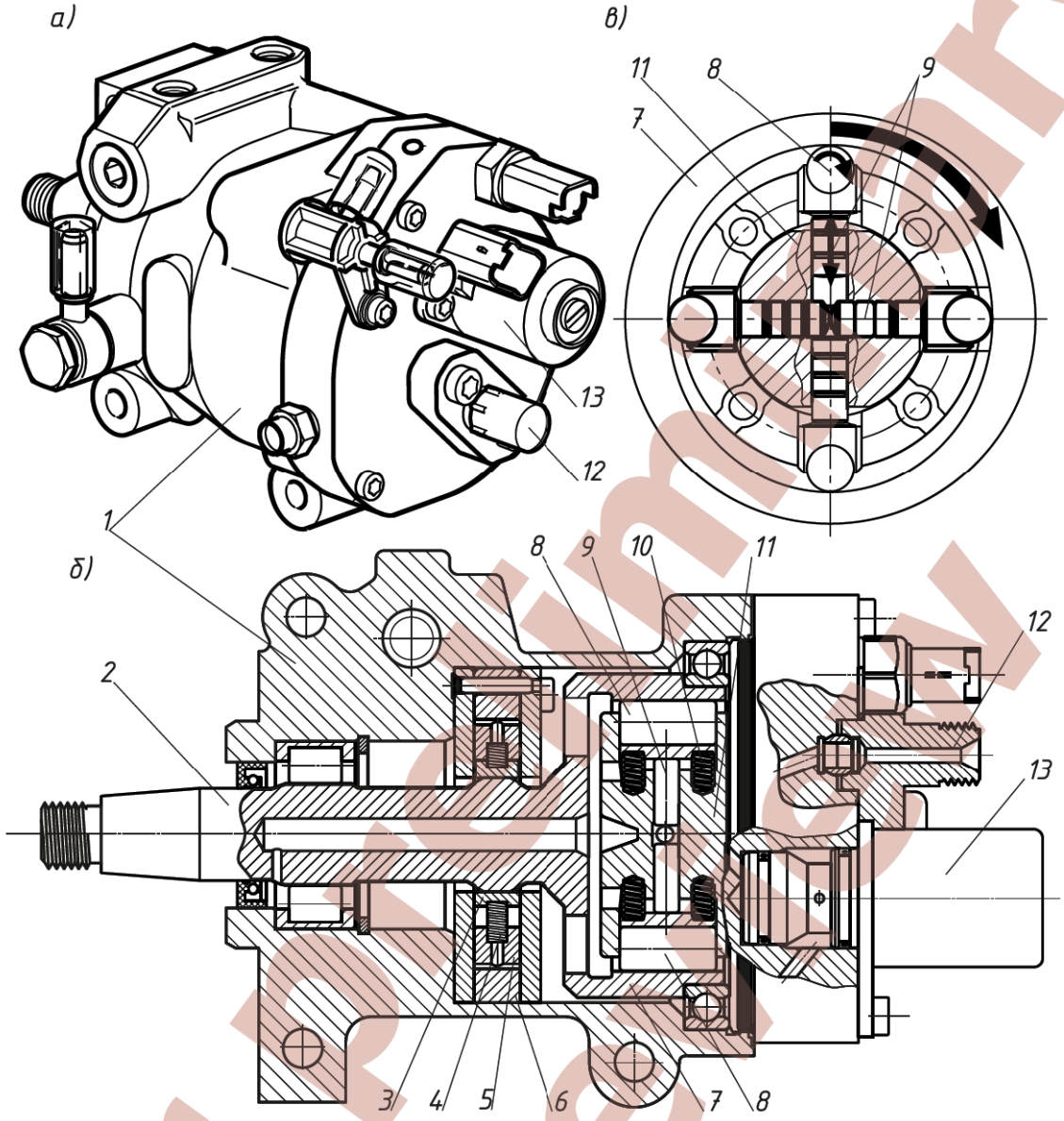


Figure 5.124 – Delphi rotary plunger pump type DFP1 for Common Rail systems of high-speed diesel engines: *a* – general view; *b* – longitudinal section; *c* – high-pressure pumping section with two pairs of plungers; 1 – pump housing; 2 – drive shaft; 3 – vane type booster pump rotor; 4 – spring for pressing the plate to the stator; 5 – pump plate; 6 – booster pump stator; 7 – rotor of the high pressure section; 8 – roller plunger pusher; 9 – plunger; 10 – plunger return spring; 11 – stator of the high pressure section; 12 – high pressure line connection fitting; 13 – fuel control valve (adapted from [76])

High pressure radial plunger pumps with opposed or star-shaped pump sections ensure uniform loading of the drive shaft, supplying fuel to the accumulator at a pressure of 120...180 MPa. The general diagram of a double plunger pump with an opposed arrangement of pump sections and the order of its operation are shown in Fig. 5.125.

Each pump section consists of a plunger pair, a suction valve and a delivery valve. In addition, there is a common pressure limiting valve at the inlet to the high-pressure module on the pumping side and a check valve that prevents emptying of the high-pressure cavities and delaying the start of the diesel engine. Lubrication and cooling of the high-pressure module is carried out by fuel, for which it is circulated through a low-pressure line.

For preliminary review

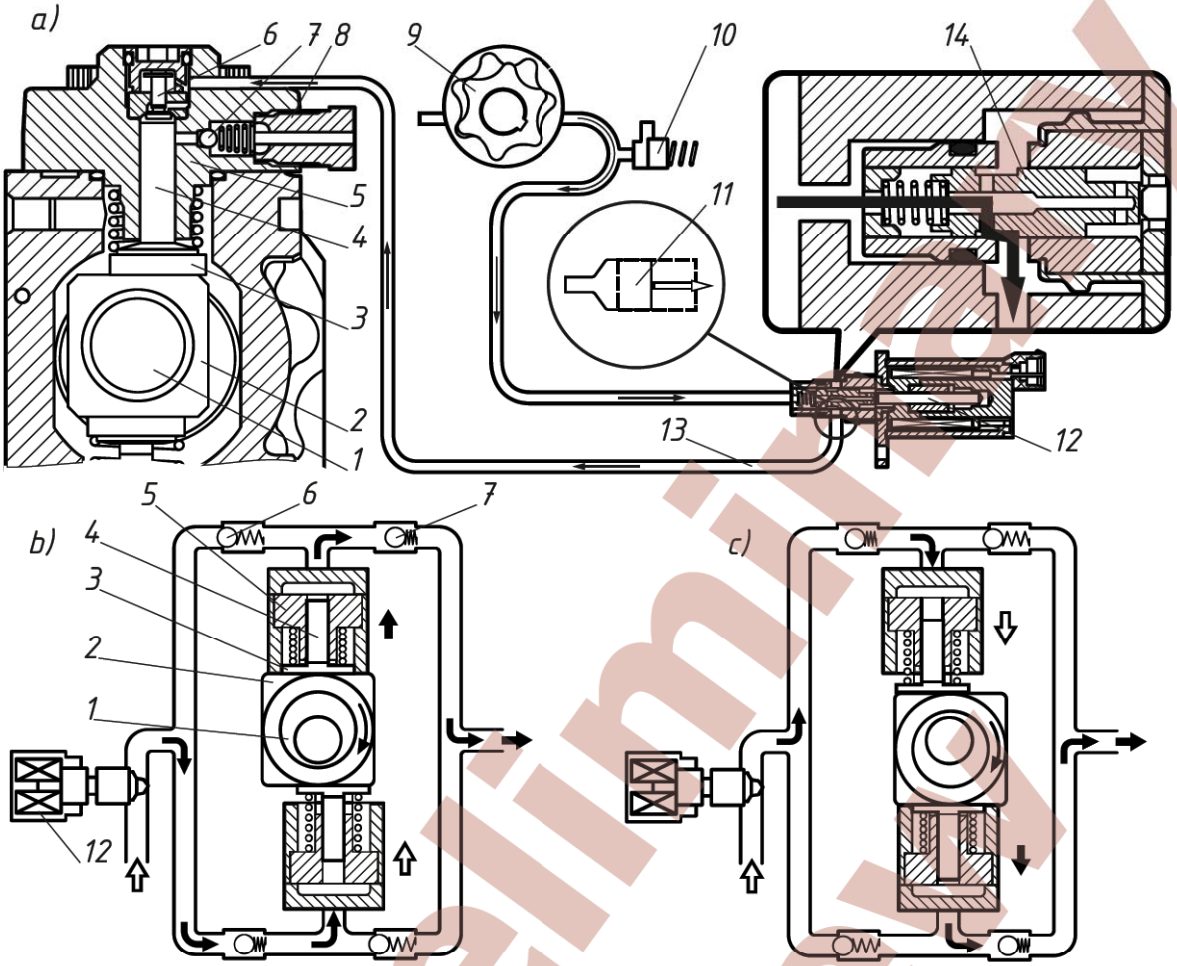


Figure 5.125 – Diagram of a double-plunger modular high-pressure pump (*a*) and its operating order: *b* – upper section during the fuel injection stroke, lower section during the filling stroke; *c* – upper section during fuel filling, lower during pumping; 1 – drive eccentric; 2 – rotor; 3 – plunger pusher; 4 – plunger; 5 – plunger bushing housing; 6 – suction valve; 7 – delivery valve; 8 – connection fitting for the discharge line; 9 – booster pump; 10 – safety valve; 11 – throttling slot of the flow control valve; 12 – flow control valve; 13 – fuel supply line to the high pressure section; 14 – spool for adjusting the fuel supply to the high pressure sections (adapted from [74])

To regulate performance, a supply control valve is installed in the pump housing, which throttles the fuel flow at the inlet to the filling line. To do this, the solenoid, under the influence of a series of pulses, moves the valve spool, which moves along the filling slot of variable cross-section (Fig. 5.125 *a*). The current value of the flow area, and therefore the amount of fuel entering the high-pressure sections, depends on the position of the valve spool, which, in turn, depends on the duration of the pulses supplied to the solenoid (pulse width modulation method). By changing the duration of the pulses, the electronic regulator controls the pump flow.

The appearance and general structure of radial plunger pumps with a star-shaped and opposed arrangement of pumping sections from various manufacturers are shown in Fig. 5.126 and 5.127.

The drive shaft of the pump with an eccentric rotor rotates in plain bearings installed in its housing and is driven by the diesel crankshaft. Moreover, due to the absence of the need to synchronize the supply, the pump rotation speed does not have to be a multiple of the rotation speed of the engine itself.

In the Bosch pump housing shown in Fig. 5.127, three plungers are located radially at an angle of 120°, and a rotor is eccentrically installed on the drive shaft in the form of a solid rotating washer with flats located at 120° intervals around the circumference of the rotor. The plunger pushers rest on the flats.

As the rotor rotates, the flat acts on the pusher and moves the plunger from BDC to TDC. The

For preliminary review

plunger returns back under the action of a spring. For one revolution of the shaft, three pump strokes are carried out.

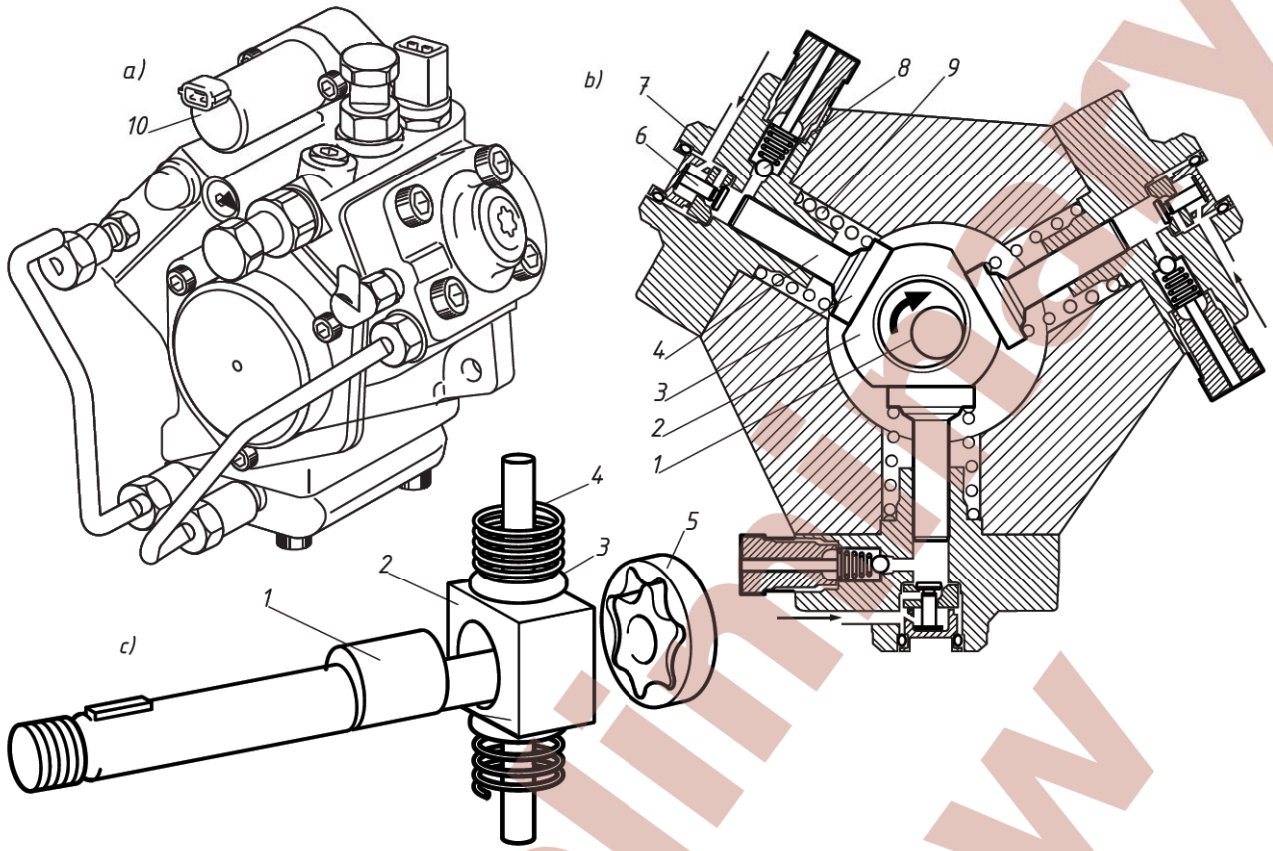


Figure 5.126 – Delphi type DFP3 plunger pump for Common Rail systems of high-speed diesel engines: *a* – general view of a three-plunger pump with pump sections located in the same plane at an angle of 120°; *b* – diagram of a three-plunger pump; *c* – diagram of a pump with an opposed arrangement of pumping sections; 1 – shaft with drive eccentric; 2 – rotor; 3 – plunger pusher; 4 – plunger; 5 – booster pump; 6 – suction valve; 7 – plunger sleeve with cover; 8 – delivery valve; 9 – return spring; 10 – flow control valve (adapted from [74])

The disadvantage of such a drive is the presence of slippage between the rotor flat and the pusher, as a result of which forces arise that distort the plunger in the sleeve and increase the power consumption of the pump drive.

When the plunger moves from BDC to TDC, the suction valve connecting the space above the plunger with the inlet cavity is closed. An increase in pressure in the above-plunger cavity leads to the opening of the delivery valve, and fuel under pressure enters the accumulator. After the plunger passes TDC, the pressure in the space above the plunger drops below the pressure in the accumulator, and the delivery valve closes. Next, when the pressure drops below the pressure in the pumping cavity, the suction valve opens and fuel begins to flow into the supra-plunger cavity until the plunger lowers to BDC. Then the process is repeated.

In a number of pumps designed for the large flow required for stable operation of the engine at full load, at low loads a large amount of fuel is discharged from the accumulator into the return line. In addition, fuel, like any liquid, heats up during compression, and prolonged operation at low loads leads to an increase in its temperature, which leads to a decrease in engine efficiency. To prevent heating and to reduce the cost of pump drive, a number of designs provide for the shutdown of one plunger section. To do this, an electromagnetic valve is installed on it, which, upon command from the electronic unit, presses the inlet valve, keeping it open. Fuel compression does not occur in this section, and pump performance decreases.

If it is necessary to obtain high fuel pump performance, it can be achieved by installing several star-shaped modules. In Fig. 5.128 shows the design of a fuel pump for high-speed high-power en-

For preliminary review

gines manufactured by L'Orange GmbH. These pumps are equipped with engines of the 4000 series from MTU, as well as engines from other companies.

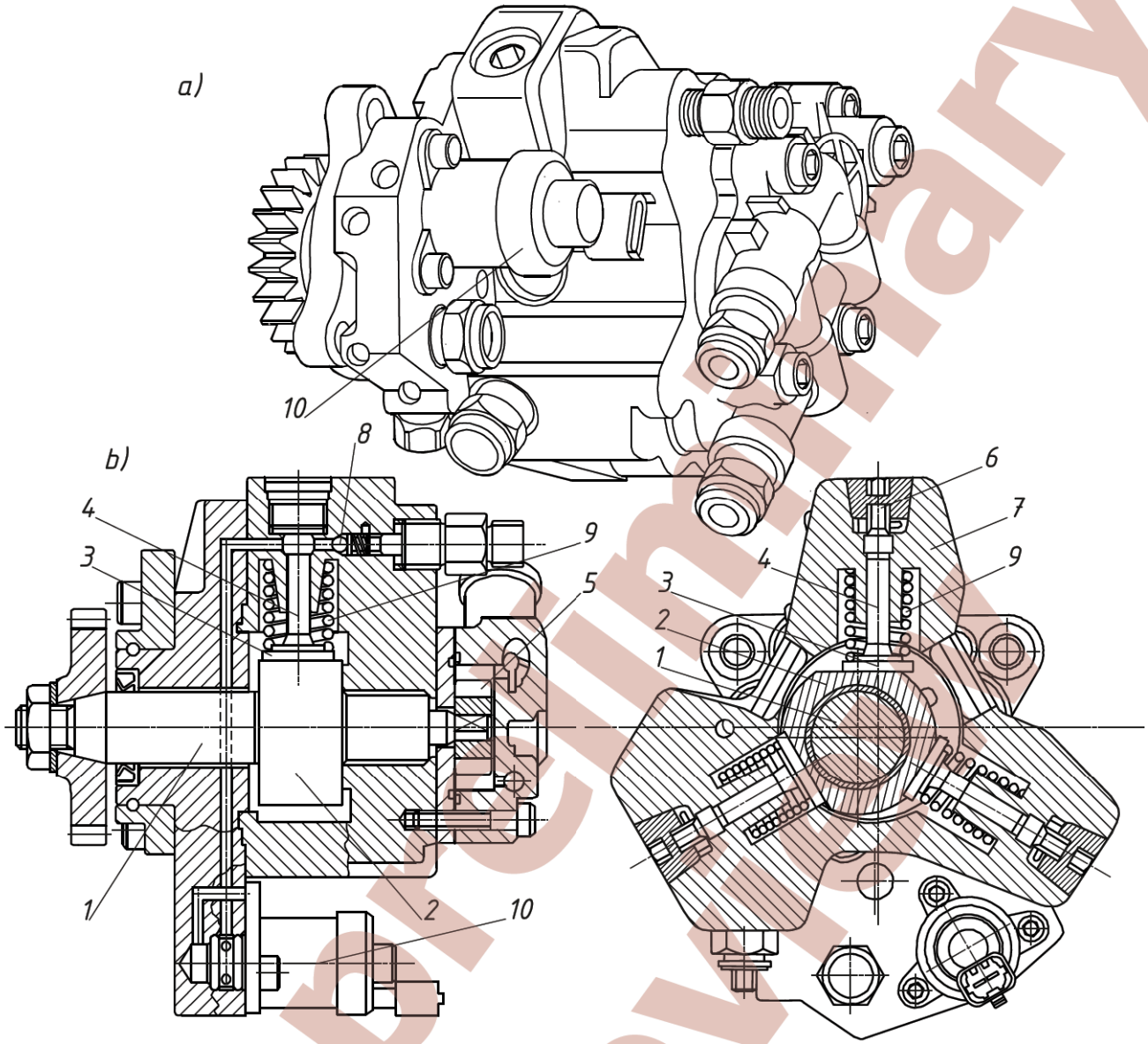


Figure 5.127 – Bosch type C3 plunger pump for Common Rail systems of high-speed diesel engines: *a* – general view of the pump; *b* – pump design; 1 – shaft with drive eccentric; 2 – rotor; 3 – plunger pusher; 4 – plunger; 5 – booster pump; 6 – suction valve; 7 – plunger sleeve with cover; 8 – delivery valve; 9 – return spring; 10 – supply control valve (adapted from [77, 78])

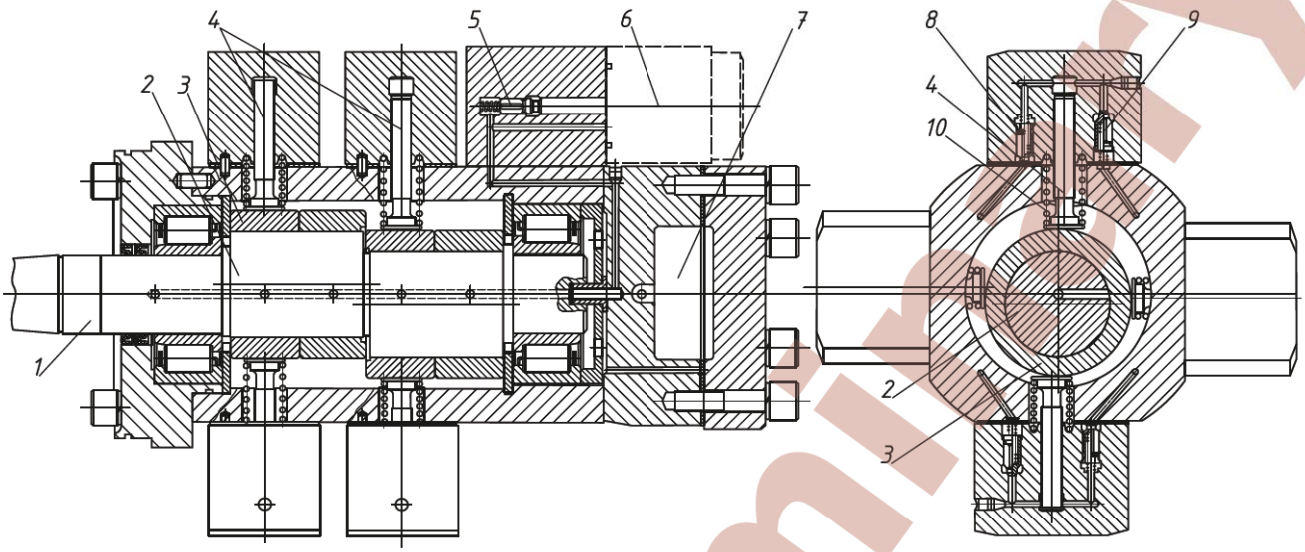
The pump is made according to a four-beam design and has eight pumping sections (two per eccentric). Unlike the previously discussed pumps, the intermediate rotor bushing is not equipped with flats for the plunger pushers, and the pump sections themselves are installed deaxially relative to the axis of rotation of the drive shaft. The kinematics of the drive is such that there is no sliding of the bushing relative to the pushers under load, and when the bushing slides at twice the speed past the unloaded pusher, a load-bearing lubricating wedge is formed. The lateral force acting on the lower plate of the pusher is determined only by the sliding friction moment of the eccentric in the bushing. The angular position of the bushing is determined by the current contact between it and one of the pushers. In this regard, no more than two, maximum three, pumping sections can be driven simultaneously from one rotating eccentric washer.

The pump performance is regulated by suction throttling using an electrically controlled spool. As in previous designs, all pump elements are lubricated with fuel.

Pumps of this series are capable of providing pressure in the accumulator up to 140 MPa with a

For preliminary review

cyclic engine flow of 330 mm^3 (275 mg), therefore they are used in diesel engines with a cylinder power of about 30...50 kW. The operating speed of the pump can reach 4000 min^{-1}; a multiplier gear can be used to drive it.



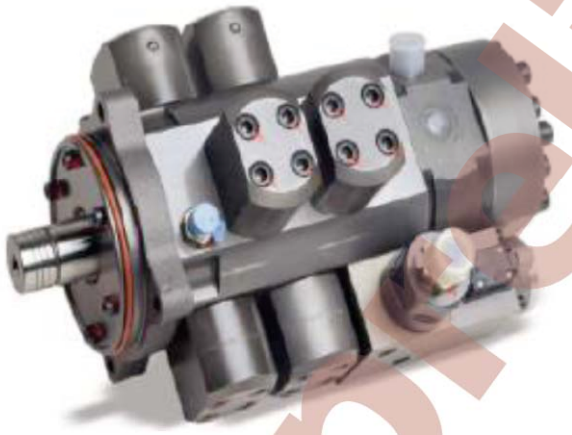


Figure 5.128 – Radial plunger pump from L'Orange GmbH for Common Rail systems of high-speed diesel engines with high cylinder power, device (*a*) and general view (*b*): 1 – pump drive shaft; 2 – drive eccentric; 3 – drive roller; 4 – plunger; 5 – flow control throttling valve; 6 – electromagnetic drive of the supply control valve; 7 – accumulator; 8 – suction valve; 9 – delivery valve; 10 – return spring (adapted from [79])

Considering that large cyclic supplys lead to significant pressure fluctuations in the accumulating cavity, when designing accumulator systems for powerful high-speed diesel engines, they try to divide the accumulator cavity into several volumes. The first accumulator cavity is provided in the pump itself (Fig. 1.128). The presence of such a cavity improves the operating conditions of automatic valves and plunger pairs of the pump.

Pressure accumulator is made in the form of a thick-walled pipeline (Fig. 5.129) with a system of threaded holes for fastening connecting fittings. To reduce the length of the discharge pipelines, the accumulator is fixed in close proximity to the injectors.

The volume of the accumulator is selected, on the one hand, for reasons of the need to smooth out pressure fluctuations from pumping sections and injectors within the permissible amplitude. On the other hand, the required accumulator volume is selected from the condition of ensuring the required speed of transient conditions. In this case, to solve the first problem, the volume must be increased, and to solve the other, it must be decreased. Thus, the choice of accumulator volume is a complex optimization problem, which is usually solved by linking the choice of volume with the dynamic performance reserve of the high-pressure pump.

Pressure fluctuations from the delivery of pump sections are less significant when using an eccentric drive due to the significantly longer pumping time of each plunger. During this period, a fuel consumption cycle occurs from at least one injector.

For preliminary review

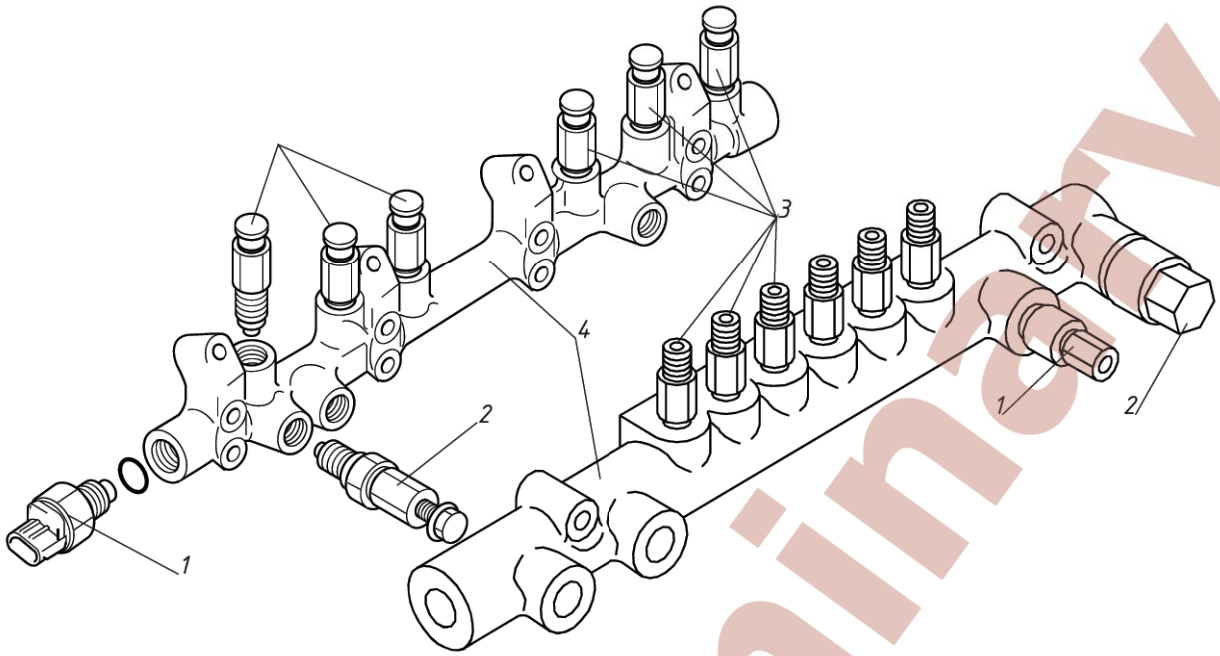


Figure 5.129 – IIigh-speed diesel pressure accumulators: 1 – pressure sensor; 2 – maximum pressure regulator; 3 – fittings for connecting high pressure lines; 4 – accumulator case (adapted from [70, 76])

The pressure control valve in the accumulator serves to stabilize the pressure when the load is suddenly removed from the engine. In this case, by reducing the flow of the high-pressure pump it is not always possible to compensate for the reduction in fuel consumption through the injectors. This can lead to an increase in pressure in the accumulator, resulting in difficulties in providing small flows, even missing individual injections. If the pressure is higher than necessary, the valve opens and excess fuel is discharged into the drain line. The general design of the Delphi control valve is shown in Fig. 5.130.

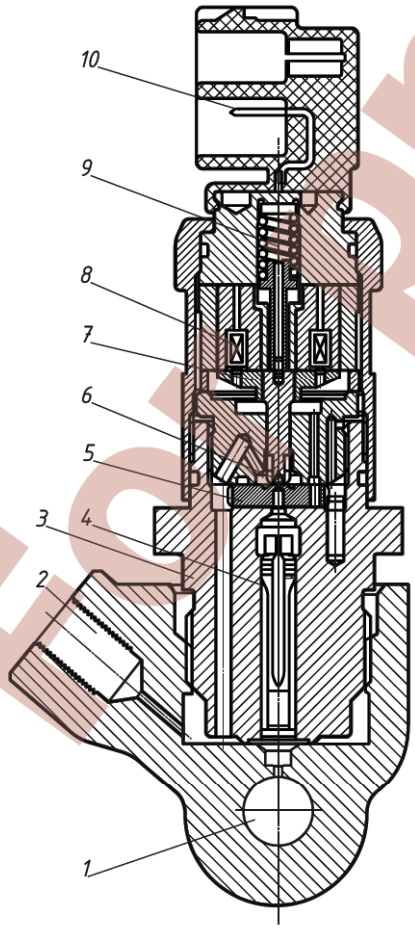


Figure 5.130 – General structure of the Delphi pressure control valve: 1 – accumulator cavity; 2 – threaded hole for connecting the drain line; 3 – valve housing; 4 – slot filter; 5 – throttle washer; 6 – ball valve; 7 – anchor; 8 – solenoid; 9 – valve loading spring; 10 – electrical connector (adapted from [74])

For preliminary review

The closing part of the valve consists of a ball, which separates the high and low pressure lines, and an armature, which, under the action of a spring, presses it to the seat. The force of the spring loading the valve is such that, under the influence of the set pressure in the accumulator, the valve separates the high and low pressure cavities. If the pressure, for example, due to wave processes in the accumulator, increases above the established norm, the valve overcomes the spring force and bleeds part of the fuel into the drain line. If it is necessary to reduce the pressure in the accumulator, voltage is applied to the solenoid, under the influence of which a current appears in the coil, creating a magnetic field that attracts the armature that holds the ball valve. Thus, this valve functions as both a limiter valve and a pressure regulator valve in the accumulator. It is assigned the role of a second control channel together with pump flow control. The pressure regulator sets the pressure in the accumulator depending on the engine load, rotation speed and thermal state of the diesel engine.

The spring stiffness is selected to limit fluctuations within 10 MPa, and the main adjustment range (23...160 MPa) is carried out using an electromagnet. The force developed by the electromagnet depends on the duration of the current pulse applied to it (pulse width modulation method). The current supplied to the electromagnet winding is constant.

It should be noted that the control method by draining fuel from a high-pressure accumulator has a number of significant disadvantages: it is too uneconomical, requires a special fuel cooler and sufficient speed of the valve itself.

For this reason, if it is possible to circumvent these shortcomings, then the non-optimality of the injection pressure, from the point of view of the requirements for organizing the operating process, is neglected in a number of designs. In such cases, the accumulator is equipped only with a maximum pressure limiting valve, an example of the design of which is shown in Fig. 5.131.

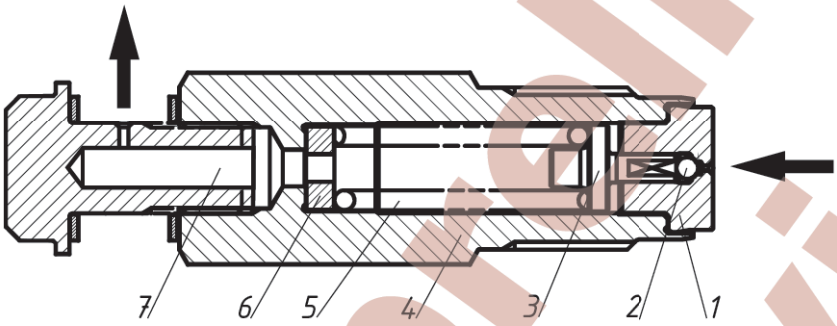


Figure 5.131 – Maximum pressure limiting valve in the accumulator: 1 – ball valve seat; 2 – ball valve; 3 – ball valve pusher; 4 – valve housing; 5 – loading spring; 6 – adjusting washer; 7 – drain line fitting (adapted from [70])

A loading spring presses the ball valve against the seat through a pusher. If the pressure in the accumulator is too high, the valve opens, overcoming the force of the spring, and part of the fuel from the accumulator is discharged through the drain line back into the supply tank.

The valve opening pressure is regulated by changing the tightening of the loading spring by selecting an adjusting washer of the appropriate thickness.

The maximum cyclic flow limiter serves to disconnect the high pressure line from the accumulator, in which depressurization occurs, for example, due to a stuck fuel injector needle or flow control valve. For high-speed engines, the flow limiter valves are usually mounted in the housing of the fitting through which the accumulator is connected to the high pressure line. The operation of the limiter is based on the principle of a pressure difference on both sides of the separating piston. In Fig. 5.132 shows the design of a ball-piston type limit valve and its operating procedure.

As fuel passes through the throttle hole in the separating piston, the cavity of the restrictor valve is filled. The start of fuel supply by the injector leads to a pressure difference between the cavity of the high pressure line and the accumulator. Since the cross-section of the throttle hole cannot compensate for this difference, the separating piston moves to the right (Fig. 5.132) and, compressing the return spring, moves the ball valve. The movement of the piston is proportional to the cyclic fuel supply, and if it does not exceed the maximum permissible value, the ball valve does not reach its extreme position, and after stopping the supply, the valve together with the piston returns to its original position. If the amount of fuel delivered by the injector exceeds the theoretical maximum

For preliminary review

flow, the piston moves the ball valve until it seats on the seat, causing the high pressure line to be disconnected from the accumulator.

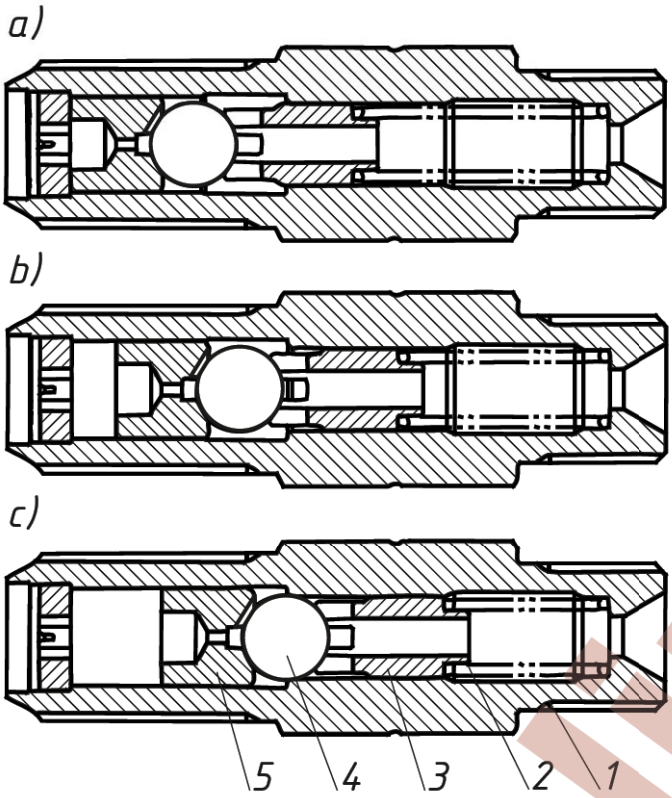


Figure 5.132 – Valve-o limiter of maximum cyclic flow: *a* – initial state of the limit valve; *b* – intermediate position corresponding to partial supply; *c* – disconnecting the high pressure line from the accumulator. 1 – valve housing; 2 – valve return spring; 3 – valve pusher; 4 – ball valve; 5 – piston with throttle hole (adapted from [76])

In addition to the emergency shutdown function, the limiting valve performs damping functions. The moving elements of the valve, having inertia, smooth out pressure fluctuations in the accumulator, ensuring fuel flow to the injectors at almost constant pressure. In addition, this valve prevents the penetration of reflected pressure waves from the high-pressure fuel injector line into the accumulator.

The diameter and length of the throttle hole are selected from the condition of guaranteed return of the valve to its original position during the period between injections at the nominal operating mode.

To monitor the pressure and temperature of the fuel in the accumulator, a temperature sensor and a pressure sensor are installed on it, the signal from which is sent to the electronic control unit. The information received from the sensors is taken into account when choosing the operating parameters of the engine fuel system.

The high-pressure pipelines that connect the accumulator to the injectors of individual cylinders are made, if possible, shorter to reduce the amplitude of the wave processes that occur in them, which accompany fuel injection. In general, the same requirements apply to pipelines as for engines with classic fuel systems. The only difference is that the pipeline is constantly under high pressure, so special attention is paid to sealing the joints.

On small diesel engines, the pipelines may not have a protective shell. In such cases, the engine is usually equipped with special shields and guards designed to limit the spread of fuel jets in the event of damage to the high pressure lines.

On high-power engines, all high-pressure pipes are placed in a protective braid, and the internal space between the main pipe and the braid is used to collect leaks.

Electrically controlled injectors for high-speed diesel engines with accumulator injection systems are structurally different from traditional injectors by the presence of a system that controls the opening of the needle valve of the nozzle. In terms of the design of housing parts and the methods of fastening in the cylinder cover, diesel injectors equipped with the Common Rail system most often do not differ from models with direct fuel injection and can be installed on basic engine models

For preliminary review

without significant modernization.

Structurally, the fuel injector consists of the following functional blocks:

- closed type nozzle with needle valve;
- pressure multiplier for the hydraulic locking system of the nozzle needle valve ;
- high-speed solenoid valve for controlling pressure in the cavity of the multiplier.

Structurally, injectors from different manufacturers may have significant differences, but the principle of their operation is approximately the same in all cases. In Fig. 5.133 shows the designs of the two most common types of high-speed diesel injectors.

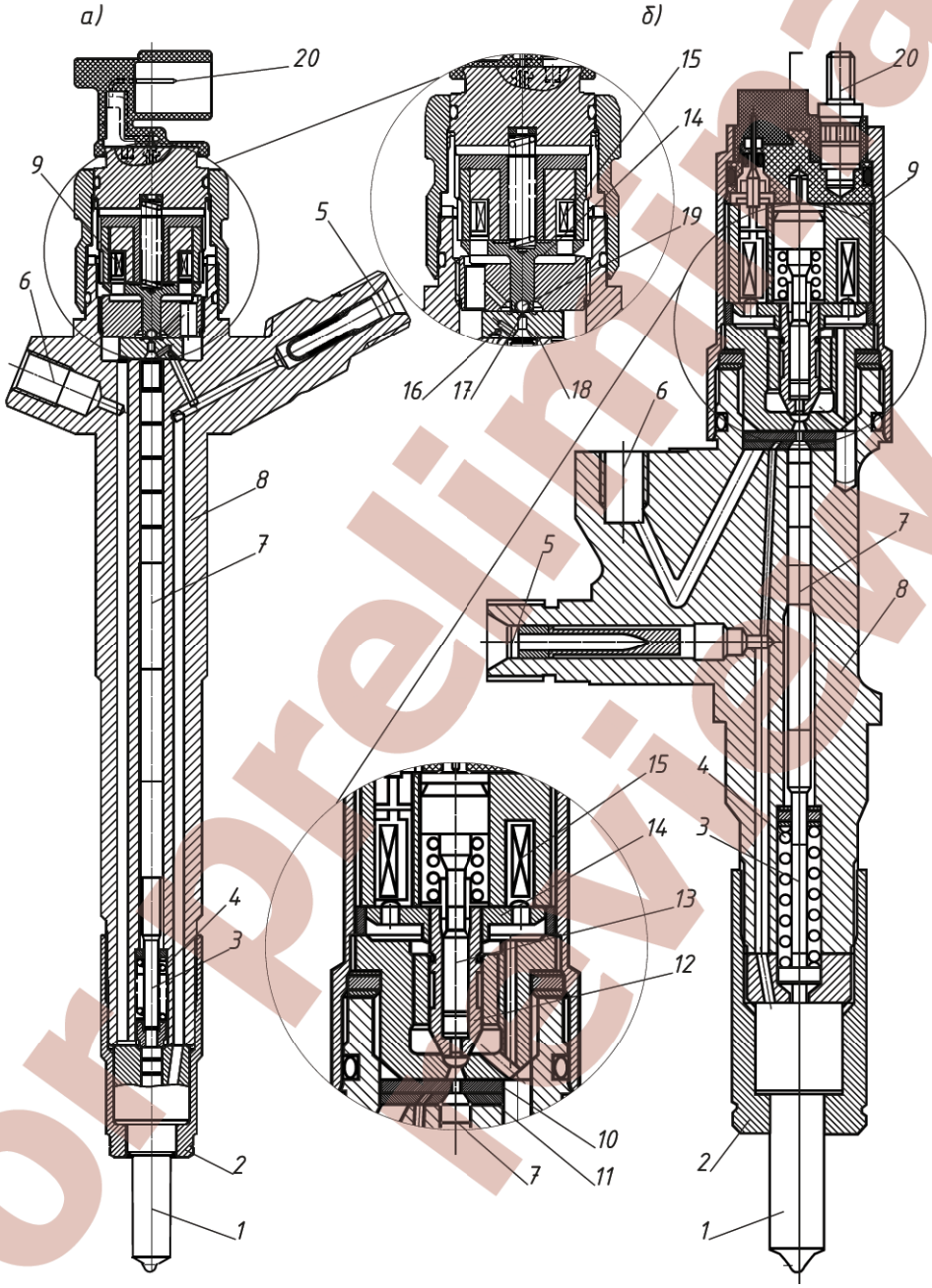


Figure 5.133 – Electrically controlled injectors of high-speed diesel engines: *a* – with a one-way solenoid valve; *b* – with a two-way solenoid valve; 1 – nozzle tip; 2 – union nut for fastening the nozzle; 3 – needle valve drive rod; 4 – needle valve return spring; 5 – fitting for supplying fuel from the accumulator; 6 – fuel drain fitting; 7 – piston of the needle valve hydraulic locking system; 8 – fuel injector housing; 9 – fuel injector control solenoid valve; 10 – upper throttle washer; 11 – lower throttle washer; 12 – internal valve; 13 – external valve; 14 – solenoid armature; 15 – solenoid coil; 16 – throttling hole on the side of the supply line; 17 – throttling hole of the pressure relief line from the hydraulic locking system of the needle valve; 18 – throttle washer; 19 – ball valve; 20 – terminal for electrical connection to the control unit (adapted from [70, 74, 76])

For preliminary review

In Fig. 5.133 *a* shows an injector made according to the most common design, with the so-called throttle control, which uses a simple single-gate valve or spool to drain fuel from the chamber of the multiplier piston of the needle hydraulic locking system. Fuel from the accumulator enters the chamber through another throttle – the filler throttle. The multiplier piston increases the locking force acting on the needle. It forms a precision pair with the housing. Its presence facilitates the layout of the fuel injector, moving the valve drive away from the nozzle, which is voluminous and intolerant of overheating.

The operating procedure of this fuel injector is shown in Fig. 5.134 *a*, *b*, *c*, and the processes occurring during the period of fuel injection by an injector are presented in Fig. 5.135.

Through channels made in the housing, fuel is supplied to the fuel injector, as well as through the throttle hole into the multiplier piston chamber. Through the fuel outlet throttle hole, which can be opened by an electromagnetic valve, the chamber is connected to the fuel return line. When the throttle hole is closed, the hydraulic force acting from above on the multiplier piston exceeds the force of fuel pressure from below on the fuel injector needle cone. As a result, the needle is pressed against the fuel injector seat and tightly closes the nozzle holes. As a result, fuel does not enter the combustion chamber (Fig. 5.134 *a*).

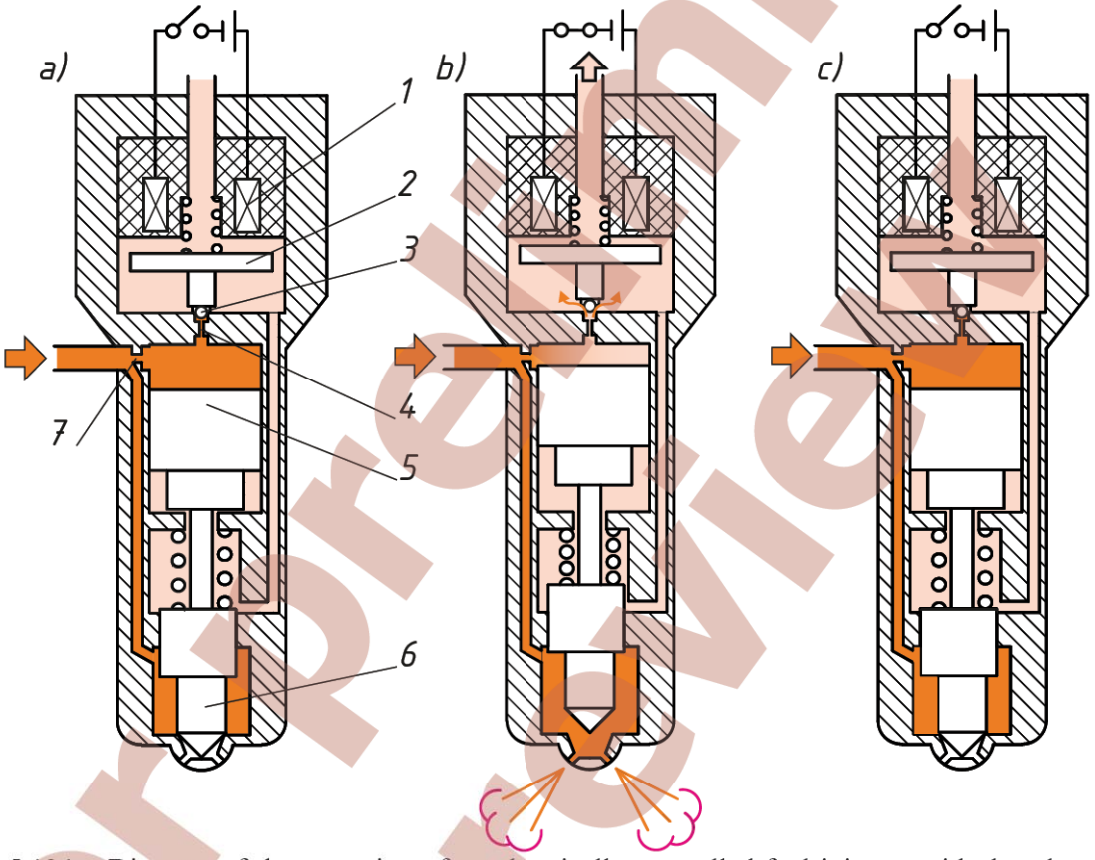


Figure 5.134 – Diagram of the operation of an electrically controlled fuel injector with throttle control: *a* – the fuel injector is closed; *b* – the injector is open, fuel is injected; *c* – stop supply; 1 – solenoid coil; 2 – solenoid armature; 3 – ball valve; 4 – throttle channel for unloading the multiplier chamber; 5 – multiplier piston; 6 – needle valve of the nozzle; 7 – filling orifice (adapted from [74])

When a control signal is supplied from the control unit (curve 1 in Fig. 5.135), the solenoid creates a force (curve 2 in Fig. 5.135), under the influence of which the armature of the control valve moves upward (curve 3 in Fig. 5.135), releasing the valve and opening the throttle hole. Accordingly, both the pressure in the chamber of the multiplier piston (curve 4 in Fig. 5.135) and the hydraulic force acting on it decrease. Under the influence of fuel pressure on the cone, the fuel injector needle comes off the seat (curve 5 in Fig. 5.135) and fuel is injected through the nozzle holes into the combustion chamber of the working cylinder (Fig. 5.134 *b*). The amount of injected fuel is proportional to the switching time of the solenoid valve and the pressure in the accumulator and does

For preliminary review

not depend on the engine speed (curve 6 in Fig. 5.135).

When the control solenoid valve is closed, all elements return to their original state (Fig. 5.134 *c*).

When the engine is not running, and there is no pressure in the accumulator, the spring presses the fuel injector needle to the seat, closing the fuel injector.

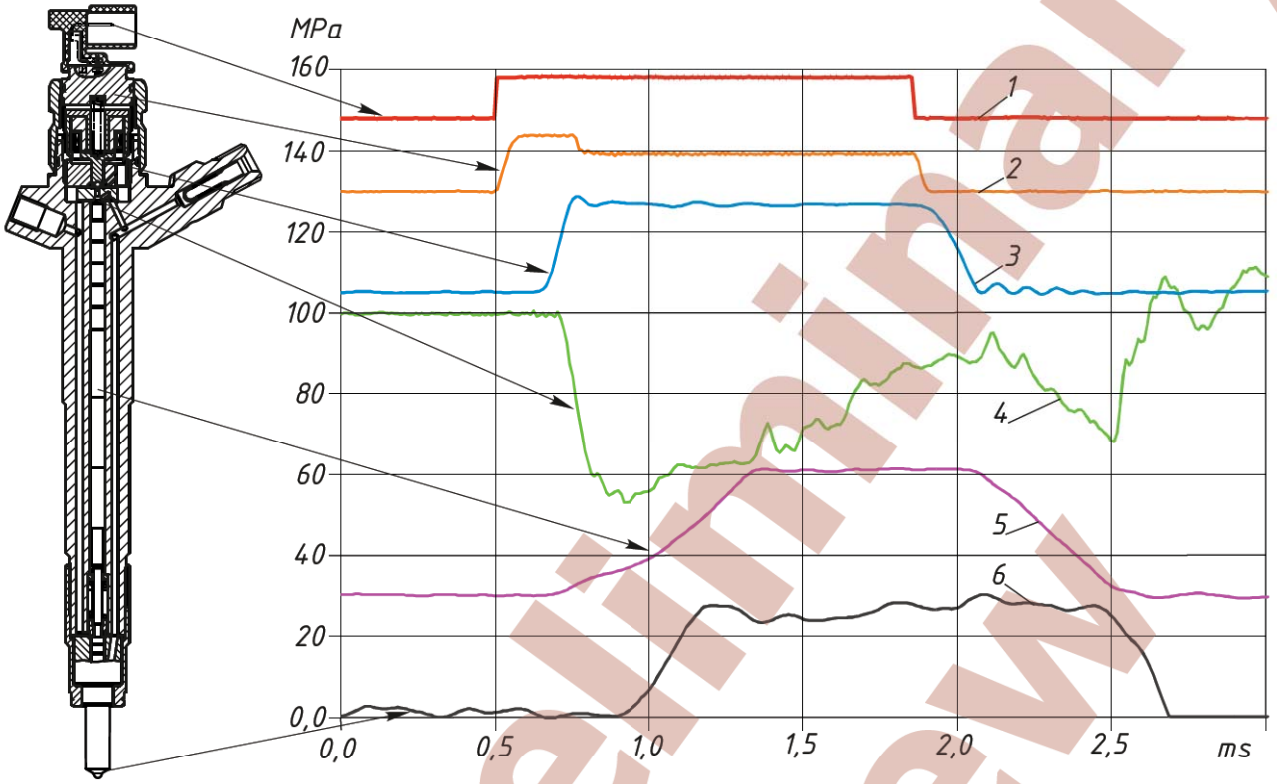


Figure 5.135 – The nature of the main processes during fuel injection using an electrohydraulic injector with throttle control: 1 – signal from the electronic control unit; 2 – solenoid force; 3 – movement of the solenoid armature; 4 – pressure in the needle hydraulic locking multiplier chamber; 5 – movement of the needle valve; 6 – fuel consumption through the fuel injector (adapted from [80])

Indirect needle control is used because the direct force of the solenoid valve is not enough to quickly raise the nozzle needle.

The disadvantage of this method of controlling fuel injection is the presence of the so-called control supply, which is an additional amount of fuel required to control the operation of the injector. After injection is completed, the fuel that controls the supply is discharged into the return line. In the process of opening the fuel injector, the multiplier piston, with its upper end, tends to block the axial throttle hole leading to the ball valve. As a consequence of this, the unloading of the multiplier chamber is reduced, but its filling continues through the fuel supply fuel injector from the accumulator. As a result, the pressure in the intensifier chamber begins to rise again, moving the intensifier piston down, and the pressure in the chamber begins to fall again. As a consequence of this, a self-oscillatory process occurs, which continues until the multiplier piston takes an equilibrium position at some distance from the stop. In this case, the minimum required fuel consumption for control is set (curve 6 in Fig. 5.135). This establishes a feedback relationship between needle lift and control fuel consumption. As a result, the response time of the needle valve and the fuel consumption for control are reduced.

In addition to the control supply, there are fuel leaks through the nozzle needle and the multiplier piston guide. All this fuel is diverted into the return line and returned to the supply tank. Other components of the injection system are also connected to the return line.

The presence of fuel consumption for control leads to the need for additional power consumption to drive high-pressure pumps. Compared to traditional fuel supply systems, this increase can be up

For preliminary review

to 50%, which, of course, reduces the overall engine performance.

To reduce fuel consumption for control, a number of high-speed engines use injectors with a hydraulically unloaded double-gate control valve. An example of the design of such a fuel injector is shown in Fig. 5.133 *b*, and the diagram of its operation is in Fig. 5.136.

When closed, fuel is supplied to the intensifier chamber through the throttle hole in the housing of the external valve. Filling the cavity inside and below the valve, fuel, through the second throttle hole, enters directly into the control chamber. The annular groove inside the outer valve coincides in diameter with its seat, therefore all forces acting on the valve are balanced, and the force of pressing the outer valve to the seat is determined by the stiffness of the return spring (Fig. 5.136 *a*). When a control signal is received, the solenoid raises the armature and the edge on the side of the internal cavity of the valve rests against the end of the fixed guide. The multiplier cavity is disconnected from the fuel supply channel from the accumulator and connected to the drain channel (Fig. 5.136 *b*). Next, the needle valve of the nozzle overcomes the force from the multiplier piston, rises, and fuel injection begins.

When the solenoid valve is de-energized, all elements of the fuel injector return to their original state (Fig. 5.136 *c*). An injector of this design has a significantly lower fuel consumption for control, the cyclic value of which is determined by the volume of the multiplier chamber.

The most difficult part of this design is the precise mating of two precision valve surfaces. This determines the scope of use of this type of injector for diesel engines with moderate speed.

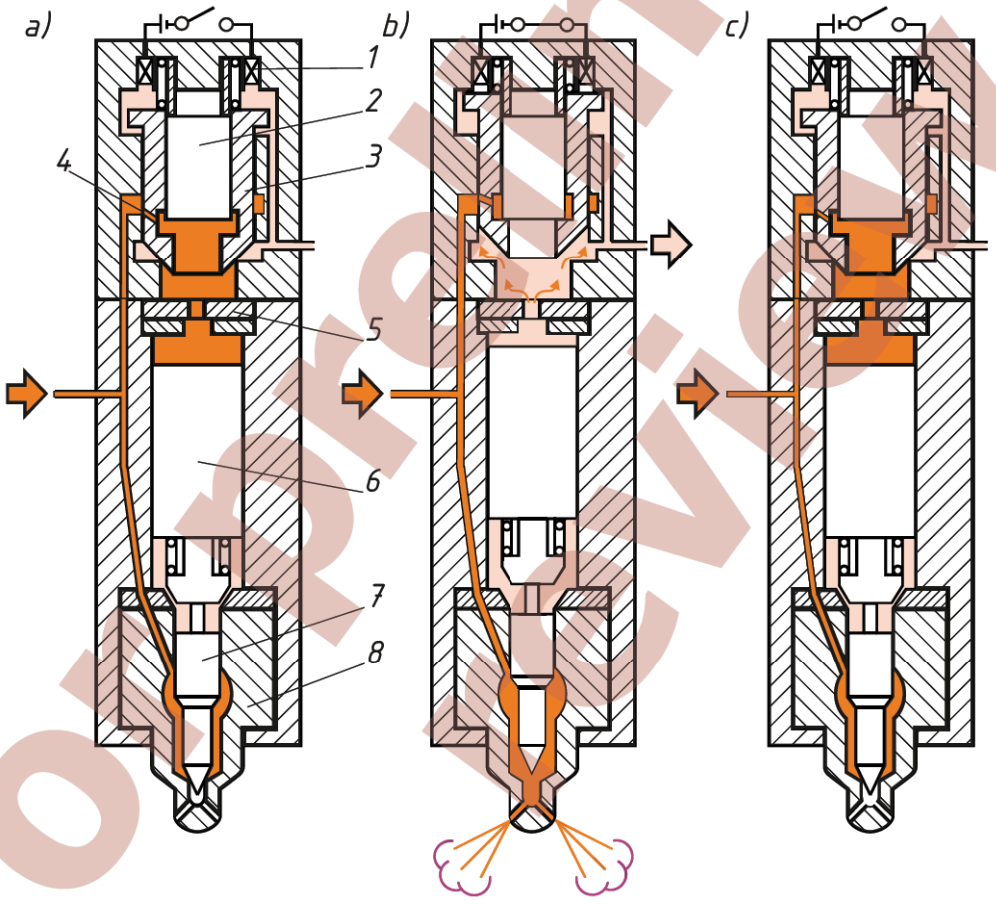


Figure 5.136 – Diagram of the operation of an electrically controlled fuel injector with a hydraulically unloaded double-shutter control valve: *a* – the fuel injector is closed; *b* – the injector is open, fuel is injected; *c* – stop supply; 1 – solenoid coil; 2 – internal valve guide; 3 – double-shutter valve; 4 – throttle channel for supplying fuel from the accumulator; 5 – throttle for unloading the animator chamber; 6 – multiplier piston; 7 – needle valve of the nozzle; 8 – nozzle housing (adapted from [76])

If it is necessary to provide large cyclic fuel supplies, a significant decrease in injection pressure may occur at the end of the injection. This is explained by accumulator discharge, when the removal of a large amount of fuel leads to a drop in pressure in the accumulator cavity.

For preliminary review

To reduce the pressure drop, the volume of the accumulator cavity must be increased, placing a significant part of it as close as possible to the nozzle. For this reason, on high-speed engines with high cylinder power, an injector with an additional accumulator chamber located in its housing is used.

An example of the design of such an injector used on high-speed engines of the 4000 series from MTU is shown in Fig. 3.137.

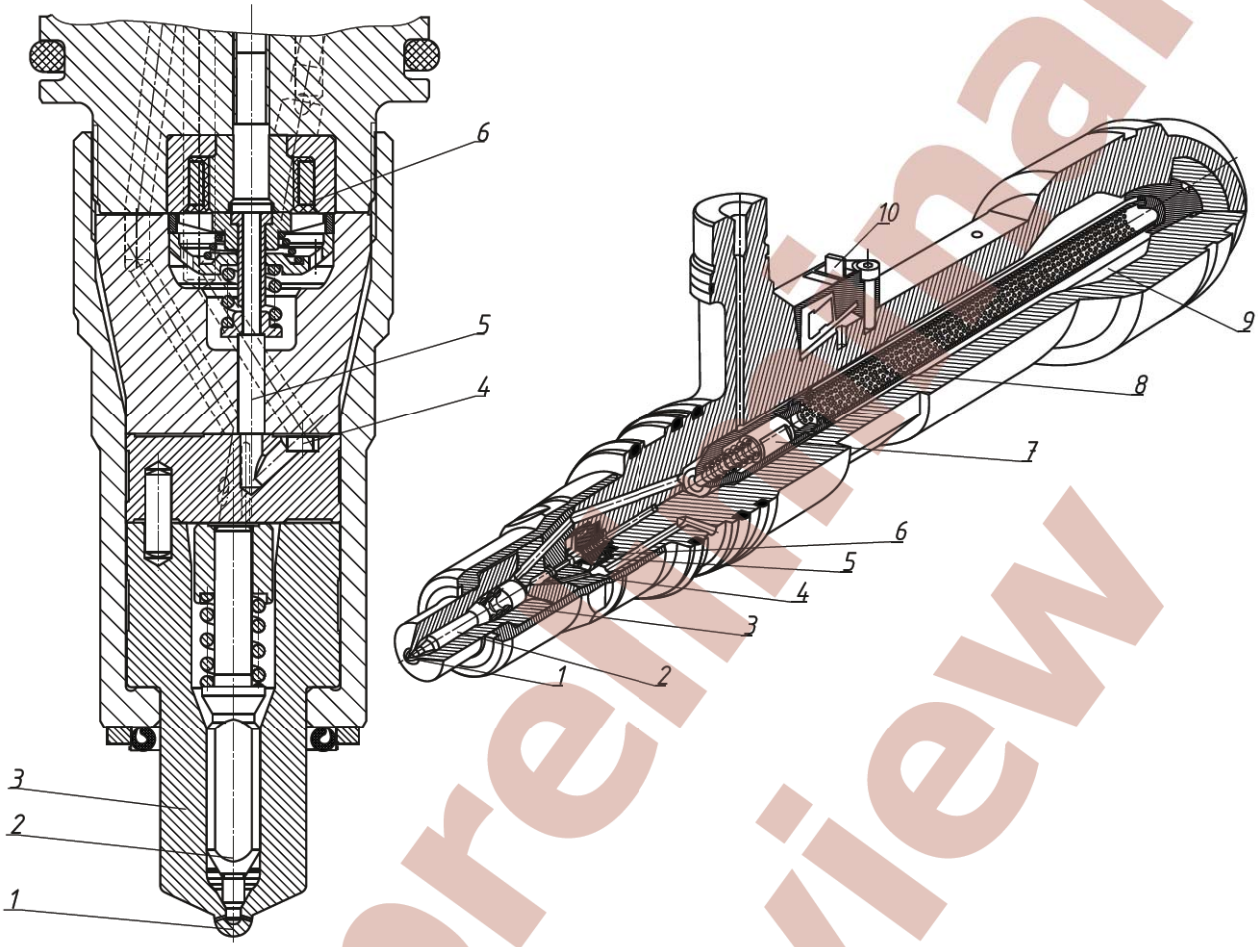


Figure 3.137 – Electrically controlled high-speed diesel injector of the 4000 series from MTU with an additional accumulator in the fuel injector housing: 1 – nozzle; 2 – needle valve; 3 – nozzle housing; 4 – channel for draining fuel from the control line; 5 – supply control valve with electric drive; 6 – solenoid coil; 7 – piston of the maximum flow limitation system; 8 – filter; 9 – accumulator cavity; 10 – electrical connector; 11 – high pressure fuel supply fitting (adapted from [9, 81])

The presence of an additional cavity allows the use of fuel pipes with a relatively small cross-section to connect the injectors to a common accumulator. As a result, pressure waves arising in the injector cavity during injection are practically not transmitted to the fuel accumulator common to all injectors, which prevents the occurrence of wave processes in the system and, therefore, contributes to more stable operation of individual injectors.

These injectors are capable of providing fuel injection at an initial pressure of up to 220 MPa, with which the company has achieved a significant reduction in emissions of particulate soot particles due to better mixture formation. In further developments of fuel systems, MTU plans to increase the pressure in front of the fuel injector to 250 MPa.

A promising type of injector for accumulator injection systems of high-speed diesel engines is the use of a piezoelectric valve actuator, which has a significantly higher operating speed.

The most significant difference from the previously discussed designs in fuel injectors of this type is the presence of a piezoelectric valve. It consists of a piezo actuator in a housing with a plug connector, a lever multiplier and a control valve that regulates the pressure in the control chamber.

For preliminary review

The principle of operation of a piezo actuator is based on the inverse piezoelectric effect, which consists in changing the size of the piezoelectric element when voltage is applied to it. The increment in the length of the piezoelectric element is directly proportional to the applied voltage. In this way, it is possible to control the increment in the length of the piezoelectric element by changing the voltage on its plates in the range from 100 to 200 V. An example of the design of an injector with a piezoelectric valve drive is shown in Fig. 5.138 *a*.

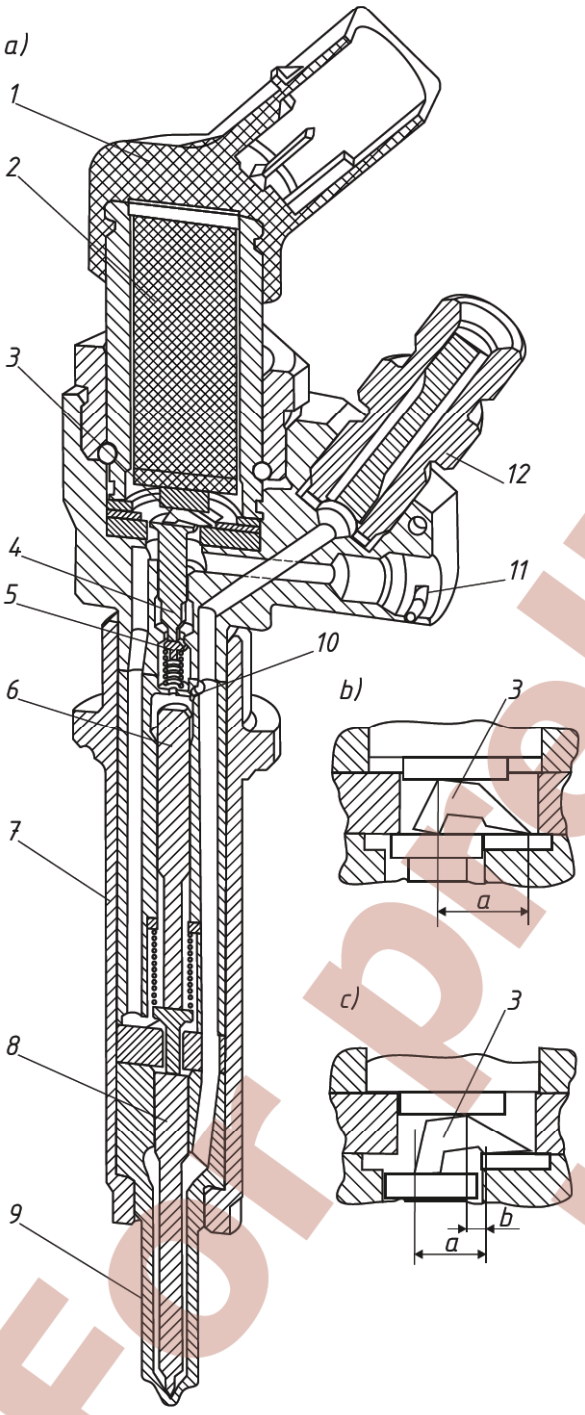


Figure 5.138 – Injector of a high-speed diesel accumulator fuel injection system with a piezoelectric control valve actuator: *a* – fuel injector design; *b* – mechanical multiplier in the position corresponding to the beginning of the valve opening; *c* – mechanical multiplier in the position corresponding to the end of the valve opening; 1 – power supply plug connector; 2 – block of piezoelements; 3 – mechanical displacement multiplier; 4 – valve drive rod; 5 – control valve; 6 – multiplier piston; 7 – fuel injector housing; 8 – needle valve of the nozzle; 9 – nozzle housing; 10 – throttle hole for feeding the multiplier chamber; 11 – drain line connector; 12 – fuel supply fitting with slot filter. *A, B* – arms of the multiplier lever at the beginning of the valve opening and at the end (adapted from [82])

The thickness of one element of the valve piezo actuator is approximately 0.08 mm, and its increase under the influence of the applied voltage is only 0.15%. Therefore, in order to obtain a displacement of order 0.04 mm, it is necessary to assemble a column from a large number of piezoelements. In a block assembled in this way, which is called a «Piezo-Stack», the individual piezoelements are separated by metal plates that serve to supply voltage to them. A column of piezoelements assembled with a pressure plate forms the basis of the piezodrive.

However, the movement in is 0.04 mm not enough for the control valve to operate normally, so

For preliminary review

its drive uses a mechanical lever displacement multiplier (Fig. 5.138 *b*, *c*).

When voltage is applied, the pressure plate of the piezoelectric element unit activates the multiplier. At the beginning of its stroke, the maximum force is transmitted through it, counteracting the high pressure of the valve opening, while the gear ratio of the drive is almost equal to unity ($a/b \approx 1$, Fig. 5.138 *b*). At the end of the stroke, the force decreases and the stroke increases ($a/b > 1$, Fig. 5.137 *c*), which ensures the valve moves by almost 0.1 mm.

A promising direction for improving accumulator fuel supply systems is the use of injectors with direct drive of a needle valve from piezoelectric elements. This solution allows you to maximize the speed of the injector, since the response speed of the piezo actuator is several orders of magnitude higher than the response speed of electromagnetic systems. In addition, the piezo actuator is capable of developing significant forces sufficient to keep the nozzle valve closed.

The main obstacle to the implementation of direct piezo drive technologies remains the relatively small movement of piezo elements, which is insufficient to ensure efficient injection of large cyclic supplys. Therefore, the scope of use of these technologies in the near future will be limited primarily to small-sized engines for ground vehicles.

As in the previously discussed MSE and HSE accumulator systems, the fuel supply for injectors with a piezo drive is controlled by an electronic control system that sets the start point and duration of injection. In this type of injectors, it is possible to vary the fuel supply law over a wider range by changing the voltage supplied to the piezo drive plates.

5.8 Fuel equipment of gas and gas-diesel marine engines

The specific operating conditions of ships have left their mark on the development of fuel systems for marine engines running on gas. This is primarily due to the need to maintain the ability to operate the engine on liquid fuels, which arises whenever the ship is moving in ballast. In addition, depending on the type of cargo, sailing conditions and time, the composition of the gases used in the power plant can change significantly, and therefore the fuel system must adequately respond to such changes and ensure the operation of the engines at nominal modes. Based on this, the bulk of marine engines today are created as dual-fuel (DF) engines, that is, capable of running on gas, liquid fuel, or both fuels at once in different proportions.

There are several fundamentally different approaches to organizing the operating process in gas engines [83]:

- converting diesel engines into engines with external mixture formation and spark ignition, operating according to the Otto cycle;
- use of external mixture formation with ignition of the gas-air mixture from a small portion of liquid fuel injected into the working cylinder;
- use of internal mixture formation and ignition of the gas-air mixture from a small portion of liquid fuel injected into the working cylinder.

Each of the above methods can be implemented using both natural and petroleum gases. All these methods have found their application in various types of marine engines, so we will consider the features of the fuel systems of each type of engine in more detail. Issues related to the storage of gases on board a ship will not be considered, since they are beyond the format of this book. This topic is discussed in sufficient detail in the relevant specialized literature.

5.8.1 Converting diesel engines to engines with external mixture formation and spark ignition

Converting diesel engines into engines with external mixture formation and spark ignition is the easiest way to convert an engine to gas fuel. Its advantages include:

- simplification of the design (if there is no need to maintain dual fuel, you can completely abandon the liquid fuel injection system, replacing it with simpler ones: an air-gas mixing system and a spark ignition system);
- operation of the supply system at low pressures, which reduces the safety requirements for the

For preliminary review

operation of such systems.

The main disadvantages of using these systems are the loss of the ability to switch to liquid fuel, as well as a decrease in liter power associated with a decrease in the weight filling of the cylinders. In addition, at compression ratios characteristic of diesel engines, detonation occurs over a wide range of loads, which prevents the use of such engines in installations with direct power transmission to the propeller. Stable operation of engines with high power and efficiency indicators is observed only in a relatively narrow range on fairly lean mixtures (Fig. 5.139). The figure shows that the optimal excess air coefficient lies in a very narrow range. When it decreases below 1.9, the engine runs with detonation, and when it increases to more than 2.2, there is a danger of misfire. This circumstance greatly complicates the qualitative regulation of engines (by changing the amount of gas supplied), therefore, in such engines, quantitative regulation by throttling the air flow at the engine inlet using an air damper is more traditional.

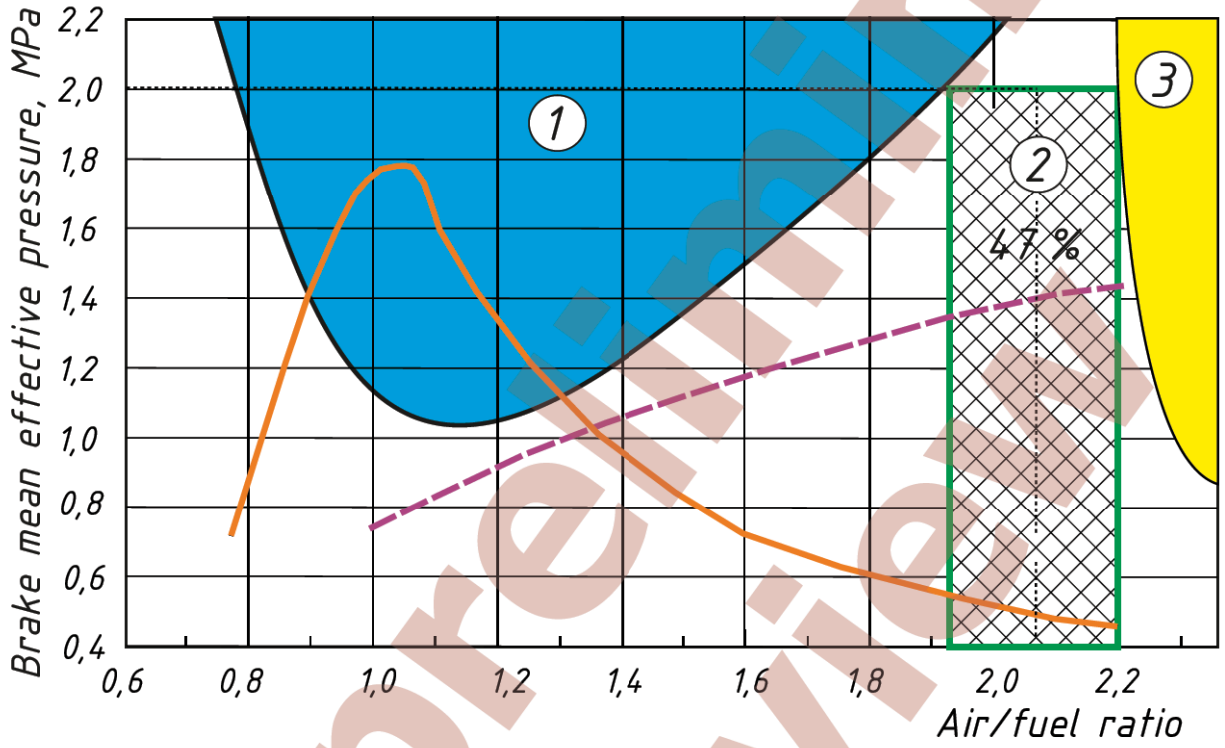


Figure 5.139 – The nature of the operating process of a gas engine operating according to the Otto cycle, depending on the excess air coefficient (based on materials from Wärtsilä): —— – NO_x does not exceed 1 g/(kWh); – – – – thermal efficiency of the engine: 1 – detonation zone, 2 – zone of optimal parameters 3 – zone of impossibility of combustion (adapted from [84])

Based on the above considerations, engines with external mixture formation and spark ignition are used mainly in power plants of drilling oil and gas production platforms, where they can be ensured of a reliable supply of gas fuel throughout the entire period of operation. Less commonly, they are used as part of multi-machine generator sets and ships with electrical power transmission to the propeller.

To obtain stable ignition and efficient combustion of lean gas mixtures, the energy of an electric spark is often not enough, so the so-called pre-chamber-plume ignition circuit shown in Fig. 5.140 is widely used in marine engines.

The prechamber is installed in place of the standard fuel injector and is a cavity connected to the main combustion chamber through several channels made in the tip of the prechamber protruding from the cylinder cover. An electric spark plug and a gas supply valve are installed in the prechamber, which is driven by the engine camshaft.

During the filling stroke, gas enters the cavity of the working cylinder through a special mixer, which distributes it in the flow of incoming air (Fig. 5.141 *a*). The gas supply from the mixer is se-

For preliminary review

lected in such a way as to prepare a lean mixture. In parallel with the flow of gas into the working cylinder through the mixer, an additional portion of gas fuel enters the cavity of the prechamber through the gas supply valve. As a result, an enriched working mixture is formed in the prechamber. When an electric spark is applied, the mixture in the prechamber easily ignites, the pressure increases sharply and burning gases in the form of plasma jets fly into the main combustion chamber, effectively igniting the lean working mixture, promoting its complete combustion (Fig. 5.141 *b*). Next, the working stroke is completed (Fig. 5.141 *c*).

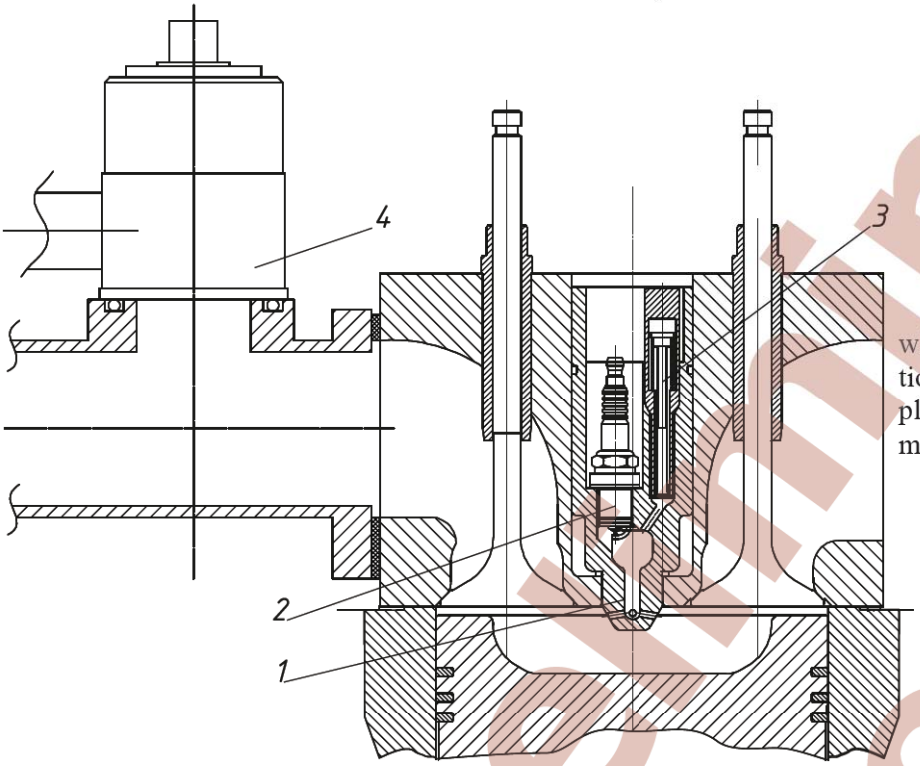


Figure 5.140 – Gas engine with prechamber-plume ignition: 1 – prechamber; 2 – spark plug; 3 – gas valve; 4 – gas mixer (adapted from [9])

A significant advantage of this method of igniting the working mixture is that the power of the ignition source in it is much greater than in a spark-ignition engine. In addition, the working mixture is ignited not at one point near the cold wall, but in the center of the charge. As a result, the efficiency of the operating processe implemented using this technology can reach 48%.

In Fig. 5.142 shows a diagram of the fuel system of an engine with prechamber-plume ignition of the KV-G3 series from Rolls-Royce. Gas is supplied to the working cylinder forcibly and is controlled using a special gas valve driven by the engine. More modern models (Wärtsilä 34SG) use electrically actuated valves controlled by an electronic unit for these purposes.

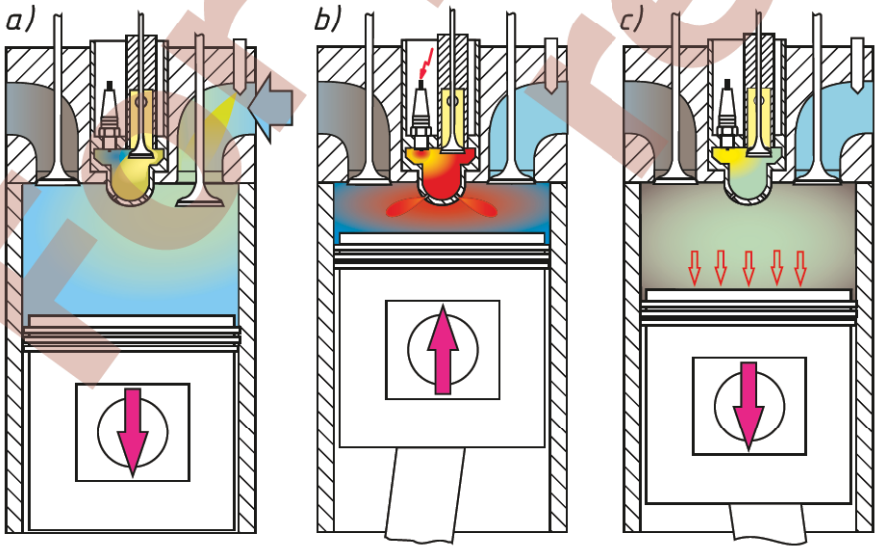


Figure 5.141 – Operating process of a gas engine with pre-chamber-plume ignition: *a* – filling the working cylinder with a lean, and the prechamber with a rich gas-air mixture; *b* – ignition of the mixture in the prechamber from an electric spark and ignition of the mixture in the combustion chamber by plasma jets; *c* – working stroke (adapted from [9])

For preliminary review

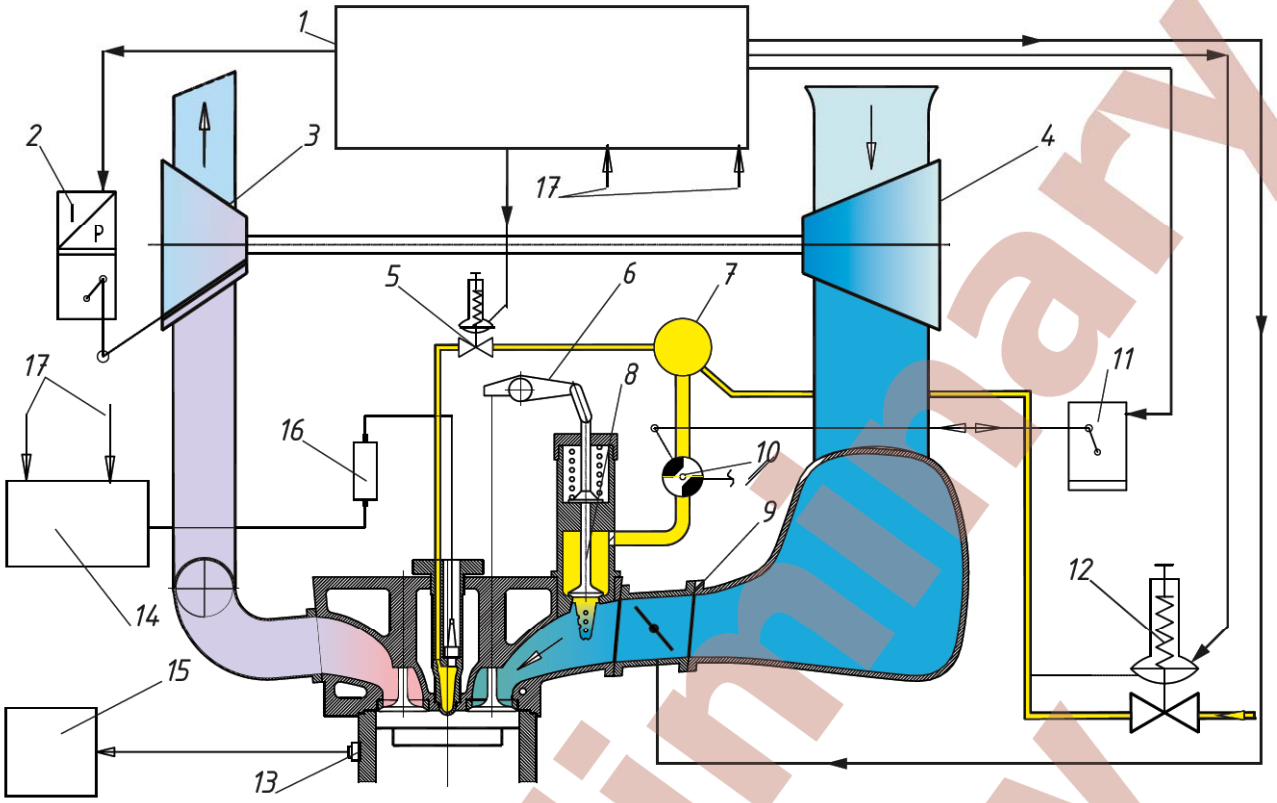


Figure 5.142 – Diagram of the gas power system for the KV-G3 engine from Rolls-Royce: 1 – engine control unit; 2 – servo drive for controlling the guide vane of the turbocharger turbine; 3 – turbine; 4 – compressor; 5 – pressure regulator in the gas supply circuit of the prechamber; 6 – gas valve drive; 7 – gas manifold; 8 – gas valve; 9 – air damper; 10 – variable throttle for regulating gas supply to the engine; 11 – servo drive of the throttle for regulating the gas supply to the engine; 12 – pressure regulator in the gas supply circuit of the working cylinder; 13 – knock sensor; 14 – spark ignition control unit; 15 – data processing unit from the knock sensor; 16 – ignition coil; 17 – inputs from sensors of rotation speed, crankshaft position, inlet air temperature, etc. (adapted from [85])

Gas mixers. To prepare the gas-air mixture at the engine inlet, various types of mixing devices are used, an example of the design of one of which is shown in Fig. 5.143. This type of mixer is used by Caterpillar in the G 3500 series gas engines.

After cleaning, the air enters the engine passing through a diffuser. As a result of a change in the flow rate, a vacuum occurs in the diffuser, which is transmitted through a special channel to the behind-the-diaphragm cavity of the control valve drive mechanism.

Under the influence of vacuum, the diaphragm bends and moves the valve, which at the end has a conical protrusion that changes the flow area of the gas supply channel. Thus, the amount of gas supplied by the mixer is always proportional to the amount of air entering the engine, thereby maintaining an optimal excess air ratio. When the engine is stopped, the lack of vacuum will cause the valve to block the flow of gas into the engine with its end surface.

Gas pressure regulator. For normal engine operation, gas must flow to the mixer at a strictly defined pressure.

The gas pressure in the storage tanks is much higher than that required for the mixer to operate, and if compressed gas is used, it can still change as fuel is consumed. Therefore, for gas engines, the issue of maintaining a constant pressure at the inlet to the engine, regardless of the pressure in the supply tanks, is acute. This problem is solved using special devices called gas reducers. Depending on the initial gas pressure, from one to three reducers can be installed in the area from the fuel tanks to the mixer. An example of a Caterpillar pressure regulator design used in G 3600 series gas engines is shown In Fig. 5.144.

This regulator is designed to work with propane-butane gas mixtures, and can also be used as a

For preliminary review

second stage of pressure reduction when using compressed natural gas. The pressure at the regulator inlet is within 2420 kPa, a t the outlet – 21 kPa.

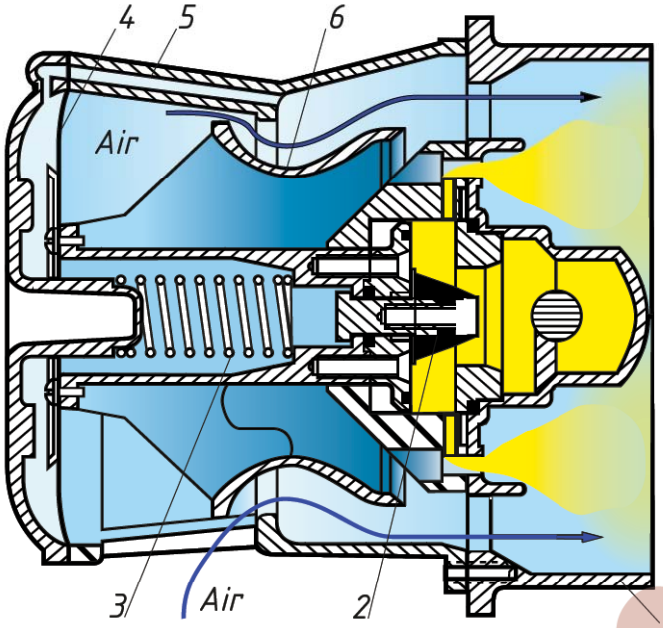


Figure 5.143 – Gas mixer for Caterpillar G 3500 series engines: 1 – mixer housing; 2 – gas valve; 3 – gas valve spring; 4 – gas valve drive diaphragm; 5 – channel for supplying vacuum under the diaphragm of the gas valve drive; 6 – diffuser (adapted from [86])

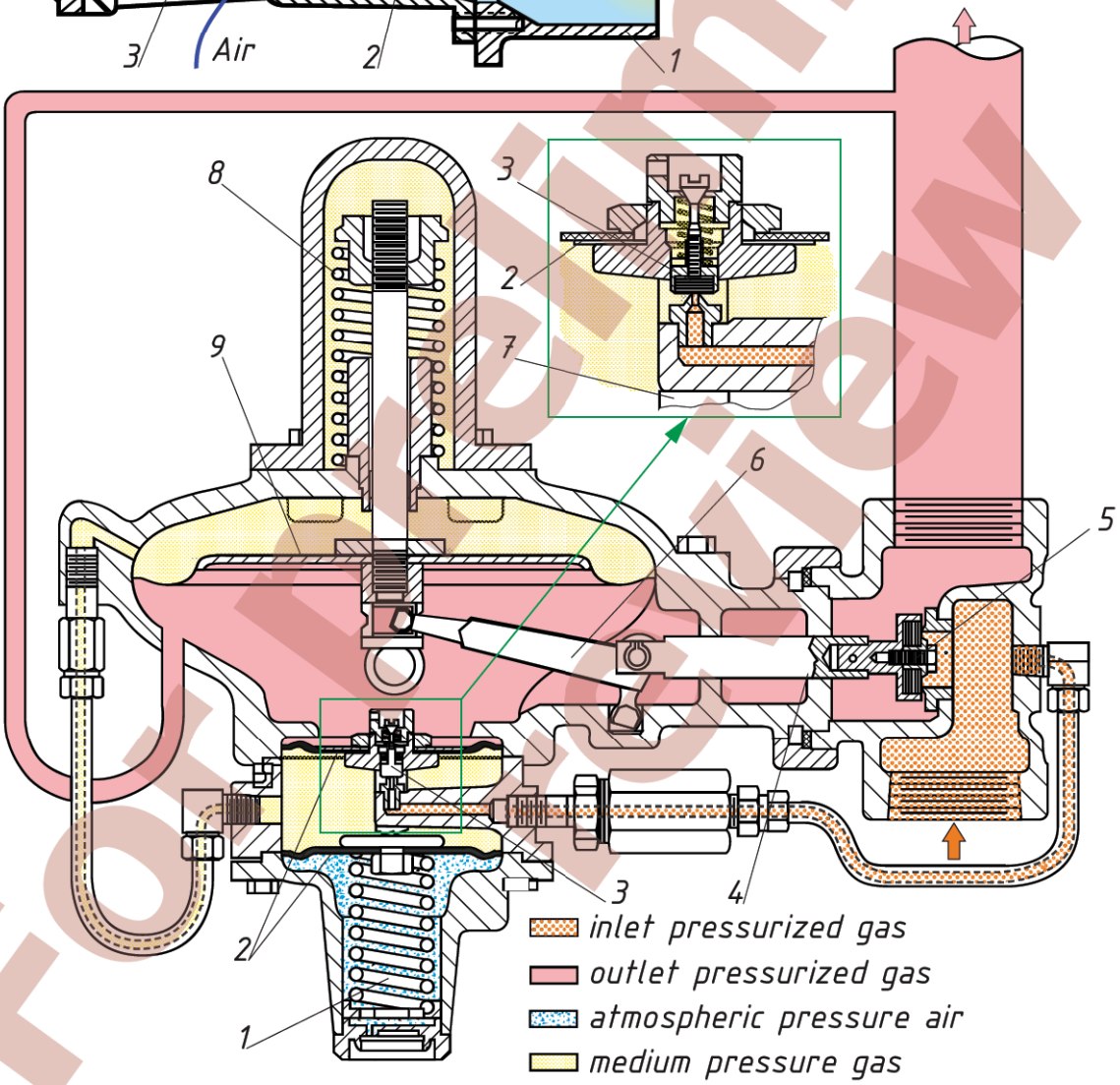


Figure 5.144 – Design and operating diagram of a dual gas reducer for a Caterpillar G 3600 series engine: 1 – operating pressure reducer spring; 2 – working pressure reducer diaphragms; 3 – double valve of the working pressure reducer; 4 – drive rod of the outlet pressure adjustment valve; 5 – outlet pressure adjustment valve; 6 – drive lever for the outlet pressure adjustment valve; 7 – intermediate stop of the operating pressure reducer valve drive; 8 – outlet pressure reducer spring; 9 – output pressure reducer diaphragm (adapted from [86])

For preliminary review

The regulator, designed to reduce pressure, is structurally a unit consisting of two reducers: working and output pressure. The gas enters the regulator after the evaporator, in which the liquid fraction is converted into gas, and also after preliminary cleaning. Initially, the gas enters the cavity of the working pressure reducer, which serves to maintain constant pressure in the working cavities of the regulator.

The working pressure reducer consists of two diaphragms, on the top of which a double valve is mounted along with its housing and seat. At the beginning of operation, in the absence of pressure, the working pressure reducer spring keeps the valve open through an intermediate stop. Therefore, when the cavity of the gearbox begins to fill, an increase in pressure causes the lower diaphragm to be pressed out, overcoming the spring force, and releases the upper diaphragm. Gas fills the cavity of the gearbox and the supra-diaphragm cavity of the output gearbox.

The output pressure reducer is used to reduce and regulate the pressure at the engine inlet. Structurally, it consists of a diaphragm loaded with a spring and a gas valve driven by the diaphragm through a system of levers.

At the beginning of operation, the spring lifts the diaphragm and, through a lever transmission, holds the valve closed – gas does not flow to the engine. When gas enters the supra-diaphragm cavity from the operating pressure reducer, the force acting on the diaphragm increases and, overcoming the spring force, the diaphragm moves down, opening the valve. Gas begins to flow to the engine. At the same time, gas fills the cavity under the diaphragm of the output reducer and creates a force tending to move the diaphragm upward, thereby closing the valve. Ultimately, a balance is established between the position of the valve, the gas flow through it and the pressure at the outlet of the regulator.

5.8.2 Fuel equipment of four-stroke gas-diesel engines with external mixture formation

The fuel equipment of four-stroke gas-diesel engines with external mixture formation is distinguished by the presence of two fuel supply systemsfor gas and liquid fuel.

The system for supplying gas fuel together with incoming air is practically no different from the previously discussed power system for gas engines with spark ignition. Considering that gas fuels have fairly low self-ignition ability, an external flame source is required to ignite the gas-air mixture. In gas-diesel engines, injection of a small portion of liquid fuel, called ignition or pilot supply, is used for these purposes (Fig. 5.145).

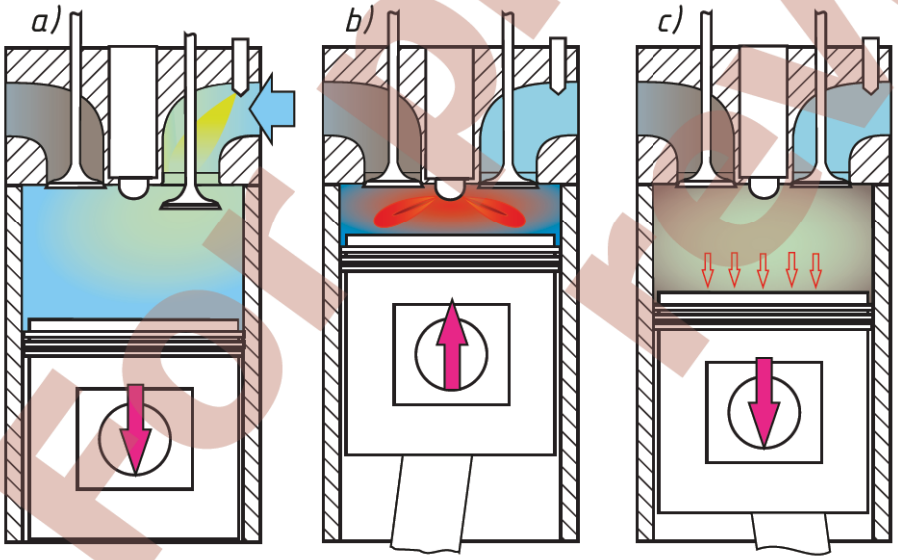


Figure 5.145 – Scheme of the operating process of a four-stroke gas-diesel engine with external mixture formation: *a* – filling the working cylinder with a lean gas-air mixture; *b* – pilot fuel injection; *c* – working stroke (adapted from [9])

There are two main approaches to organizing the operating process of a gas-diesel engine with external mixture formation:

- supply of the pilot portion using standard engine fuel equipment;
- creation of special fuel systems for injection of a pilot portion of fuel.

In the first case, the diesel engine is retrofitted with gas system components without significant

For preliminary review

modification of the standard power supply system. When operating on gas, the fuel pumps are switched to the minimum stable supply mode, and the engine load is regulated by changing the amount of gas entering the working cylinder (qualitative regulation) or the gas-air mixture (quantitative regulation).

The undoubted advantages of the first approach are: the need for minimal modification of the engine itself and its fuel system, the ability to switch to working on liquid fuel at any time. The main disadvantage of this approach is the impossibility of significantly reducing liquid fuel consumption for organizing pilot injection. Typically, in engines of this type, the ignition supply consumption is 15...20% of the liquid fuel consumption. This is explained by the impossibility of obtaining stable injection at low flow rates using standard fuel equipment. The standard fuel equipment is regulated, as a rule, to the nominal operating mode of the engine; as the cycle portion decreases, the uniformity of supply deteriorates. In addition, a decrease in cyclic supply leads to a decrease in injection pressure (Fig. 5.65), which worsens the quality of fuel atomization and can lead to misfires. A simple readjustment of the fuel equipment, as a rule, does not give the desired result, since this leads to a deterioration in the performance of the liquid fuel engine.

The second approach comes down to retrofitting the engine with an additional fuel system specifically designed to supply a pilot portion of fuel. An example of this approach is the series of gas engines from MAN, Wärtsilä, etc., created on the basis of the existing model range of diesel engines.

In some engines, to compensate for power losses when switching to gas fuel, manufacturers increased the cylinder diameter. For example, in the Wärtsilä 50DF engine, created on the basis of a 46 series diesel engine, the cylinder diameter is increased from 460 to 500 mm, and in the MAN 51/60, created on the basis of the MAN 48/60, the cylinder diameter is increased from 480 to 510 mm.

All engines are designed for continuous operation on gas, but can use MDO liquid fuel as a backup. Operation on heavy fuels such as HFO is also possible, but this requires certain preparation and the use of more expensive circulating oil.

When running on liquid fuel, the engine uses the standard fuel system. To inject the pilot portion, MAN installs an additional fuel injector, and Wärtsilä installs a dual fuel injector with two nozzles tips for injection of the main and pilot fuel.

To prepare the gas-air mixture, engines are equipped with a gas supply system to mixers installed on each cylinder cover. Thus, the preparation of the gas-air mixture is carried out directly at the entrance to the cylinder (Fig. 5.146).

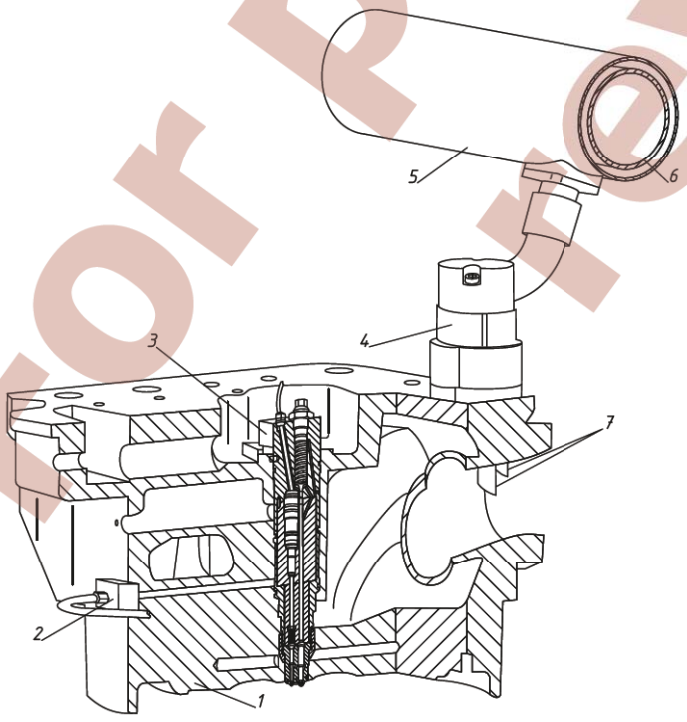


Figure 5.146 – Cylinder cover of a gas-diesel engine 20DF from Wärtsilä with a gas supply control valve: 1 – cylinder cover; 2 – fuel supply to the pilot injection system; 3 – double fuel injector; 4 – gas supply control valve; 5 – outer casing of the gas line; 6 – gas line; 7 – mixer (adapted from [87])

For Preliminary review

This solution avoids the danger of an explosion in the intake receiver, which has a fairly large volume and is often designed as a cavity in the crankcase. Therefore, a gas explosion in the receiver can lead to destruction of the entire engine.

To supply the pilot portion, modern gas-diesel engines use a accumulator fuel injection system, which turns on only when the engine is running on gas.

As noted earlier, accumulator systems provide stable fuel delivery and atomization quality regardless of engine operating mode. And the parameters of all elements of the fuel system are specially optimized to ensure a stable supply of small cyclic portions. As a result, the amount of pilot fuel was reduced to 1...5%.

The diagram of the fuel system for reserve and ignition fuel of the Wärtsilä 50DF gas-diesel engine is shown in Fig. 5.147.

To supply fuel under high pressure to the accumulator injection system, a radial-axial type fuel pump from L'Orange GmbH is used (Fig. 5.128), which is driven by an electric motor. The electric motor, like all other processes, is controlled by an electronic control unit.

After the pump, the fuel enters under a pressure of 90 MPa into an accumulator common to all injectors, from where it is supplied through separate pipelines to the injectors with two tips.

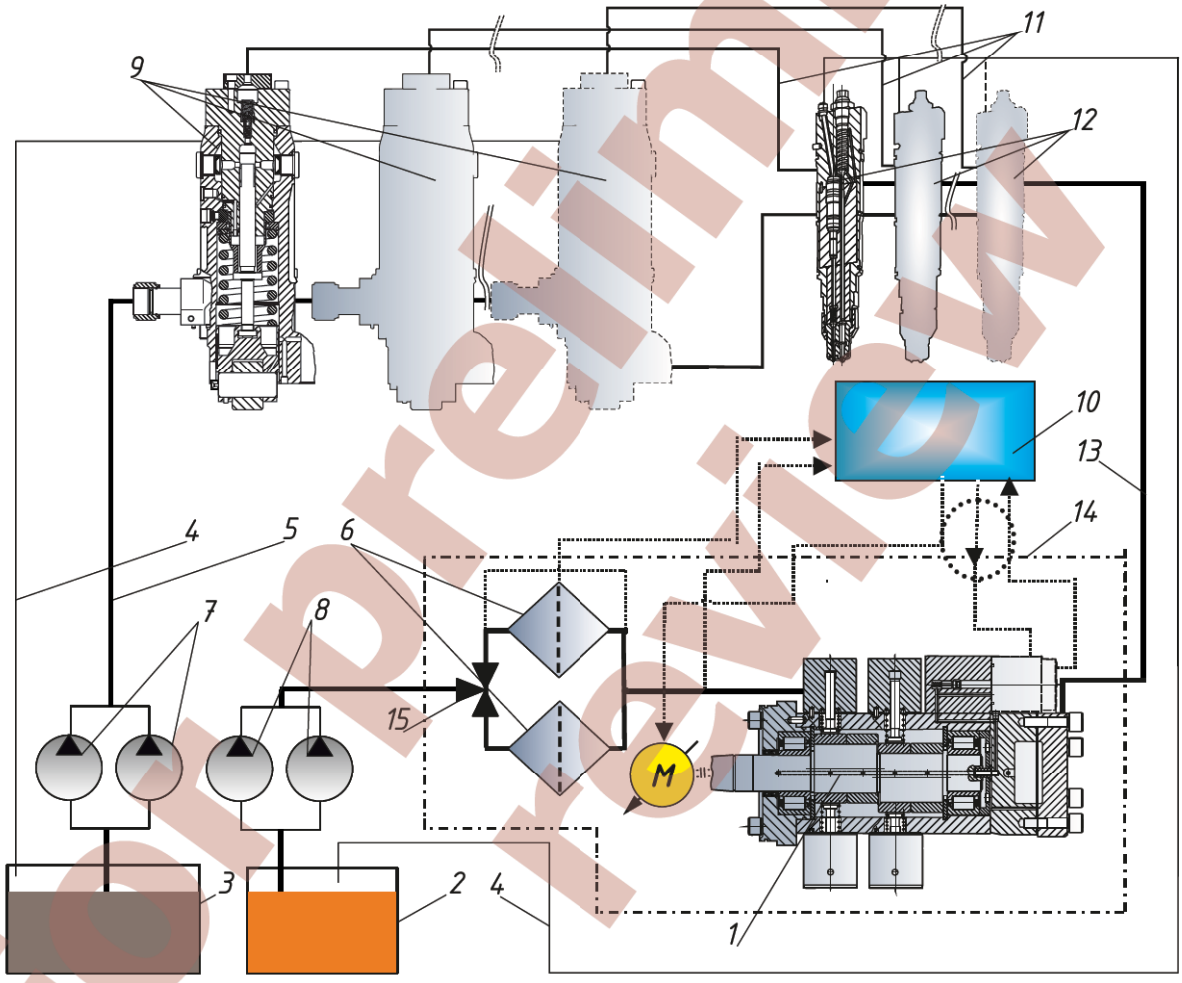


Figure 5.147 – Diagram of the fuel supply system of Wärtsilä gas-diesel engines with an additional circuit for pilot fuel injection: 1 – fuel injection pump of the pilot injection system with an electric drive; 2 – light fuel tank for pilot injection; 3 – main reserve heavy fuel tank; 4 – drain line; 5 – heavy fuel pressure line; 6 – fuel filters; 7 – heavy fuel booster pump; 8 – light fuel booster pump; 9 – standard heavy fuel injection pumps; 10 – electronic control unit; 11 – high pressure pipelines; 12 – fuel injectors with dual fuel tips; 13 – pilot fuel high pressure line; 14 – fuel pump block of the pilot injection system (adapted from [88])

The fuel injector design is shown in Fig. 5.148. In fact, there are two injectors located in one housing, one of which has a design traditional for engines with a positive displacement injection

For preliminary review

system, and the other is equipped with a solenoid valve for controlling the flow, typical for injectors in accumulator systems. The nozzle housing also has two needle valves: a small one for pilot fuel injection and a large one for main fuel injection. The pilot injection is controlled by the electronic engine control unit.

The transition from gas to diesel fuel is also controlled by the control unit. If necessary, you can switch the engine from gas to diesel fuel without stopping it at full load. The reverse transition can be carried out at a load of up to 80% of the nominal load.

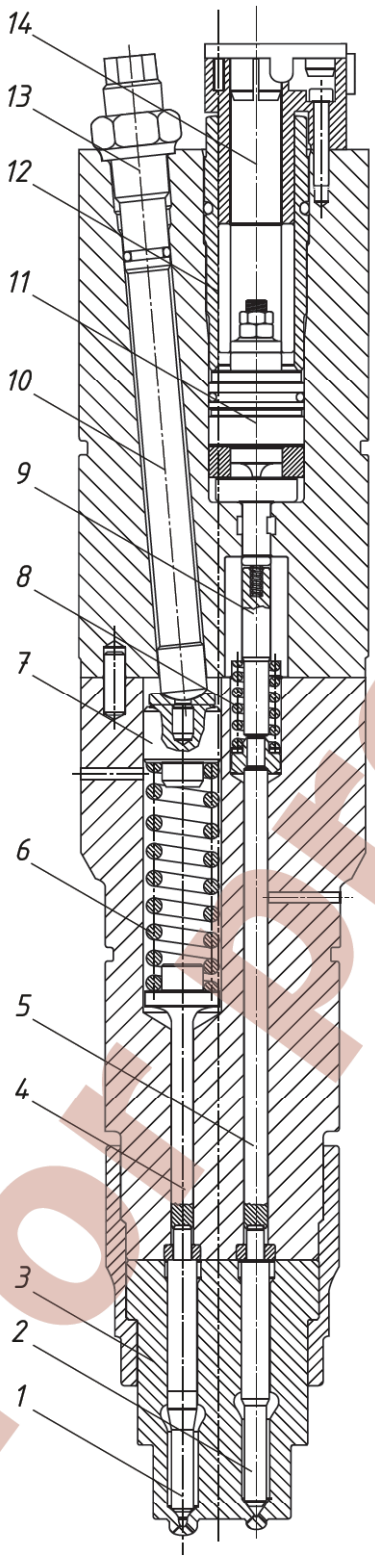


Figure 5.148 – Injector of a gas-diesel engine 50DF from Wärtsilä with a dual fuel injector for reserve and pilot liquid fuel: 1 – needle valve for supplying reserve fuel; 2 – pilot fuel supply needle valve; 3 – nozzle housing; 4 – reserve fuel needle valve rod; 5 – pilot fuel needle valve rod; 6 – main fuel needle valve spring; 7 – spring stop; 8 – spring of the ignition fuel needle valve; 9 – piston for locking the pilot fuel needle valve; 10 – spring adjusting rod for setting the opening pressure of the reserve fuel valve; 11 – pilot fuel nozzle control solenoid; 12 – installation cup; 13 – adjusting bolt; 14 – installation stop with electrical connector (adapted from [89])

The gas supply system is controlled by an electronic unit. On gas-diesel engines for marine use, the gas supply system diagram shown in Fig. 1 is widely used. 5.149.

For preliminary review

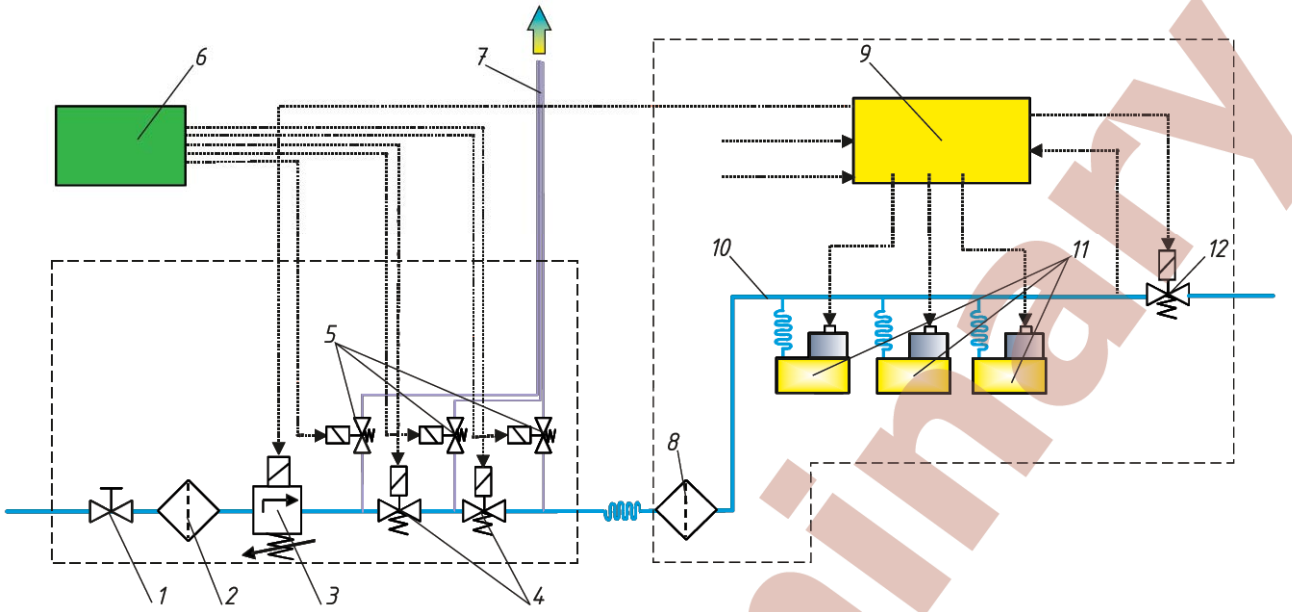


Figure 5.149 – Diagram of the gas supply system for Wärtsilä gas-diesel engines: 1 – main gas valve; 2, 8 – filter; 3 – gas pressure regulator; 4 – electrically controlled safety valves; 5 – electrically controlled safety valves of the ventilation system; 6 – electronic control unit for the gas installation; 7 – ventilation pipes; 9 – electronic engine control unit; 10 – gas main; 11 – valve for controlling the gas supply to the engine; 12 – emergency gas release valve (adapted from [88])

The gas pressure in the fuel system depends on the engine load and the calorific value of the gas, but in all cases it does not exceed 0.4 MPa. To prevent gas leaks into the engine room, all gas lines are placed in protective casings, the interior of which is constantly ventilated. To prevent mechanical damage to the supply control valves, they are connected to the main gas line using sealed bellows. At the outlet of the ventilation system, flame arresters and sensors for methane content in ventilation gases are installed. If the presence of methane is detected in the latter, the automatic protection system turns on the alarm, and if the concentration reaches 60% of the lower flammability limit of the gas-air mixture, an emergency shutdown of the engine occurs.

In the event of damage to the gas system, the engine is automatically switched to liquid fuel, and the gas is displaced using nitrogen supplied from the inert gas system. Remaining gas fuel is removed through an emergency release valve into a special combustion device.

To avoid gas fuel entering the crankcase space through leaks in the piston rings, engines are started only on liquid fuel. When maneuvering the vessel, the engine must also be switched to liquid fuel. The crankcase valves must be adjusted to take into account the possibility of an explosion in the crankcase of the gas-air mixture resulting from gas leaking through the piston rings. The crankcase ventilation system must provide effective removal of crankcase gases. It is required to be equipped with a methane sensor and an oil mist sensor. On most gas-diesel engines, the installation of heating sensors for crankshaft bearings and a crankcase ventilation system with inert gases is provided as a separate option.

The exhaust pipe of a gas engine must not be connected to the exhaust pipes of other engines.

The engine is controlled by dosing fuel using electrically controlled gas valves installed at the inlet of each engine cylinder (Fig. 5. 150).

The valve assembly consists of a housing that houses a valve in the shape of a flat cylindrical plate and a seat with a system of axial holes for the passage of gas fuel. The valve is connected to the armature of the solenoid, which, when voltage is applied to the coil, attracts the armature and opens the valve. The valve closes under the action of the return spring and gas pressure in the above-valve cavity.

Dosing of gas fuel is carried out by changing the opening time of the metering valve for each cylinder individually.

For preliminary review

Since when operating on gas, minor deviations from the set mode can lead to detonation, a knock sensor is installed on each cylinder, which is connected to the engine control unit.

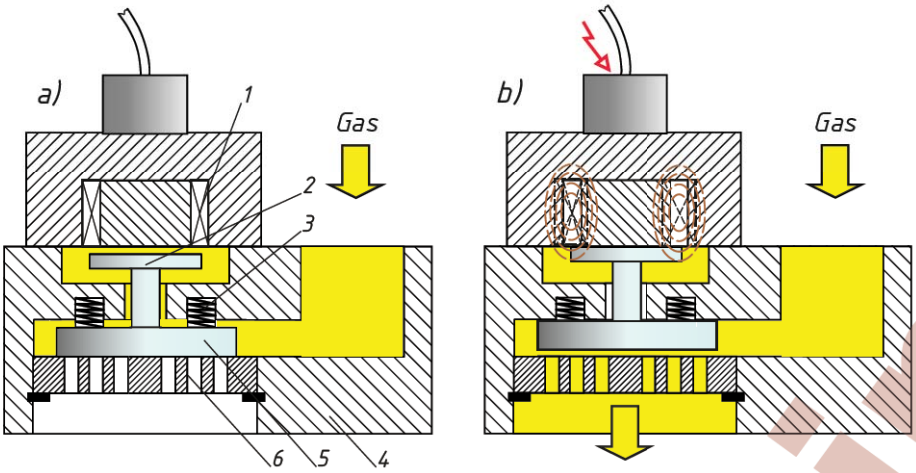


Figure 5.150 – Gas supply dosing valve in closed (*a*) and open (*b*) states. 1 – solenoid coil; 2 – solenoid armature; 3 – return springs; 4 – valve housing; 5 – valve plate; 6 – valve seat (adapted from [90])

5.8.3 Fuel equipment of two-stroke gas-diesel engines

In two-stroke engines, it is quite difficult to organize external mixture formation, since before entering the working cylinder, air fills the sub-piston cavity, which has a fairly large volume. The presence of such a large amount of gas-air mixture increases the risk of explosion and the severity of its consequences. Therefore, modern two-stroke engines use internal mixture formation, in which gas fuel is supplied to the working cylinder after the gas distribution elements are closed.

There are two main approaches to internal mixture formation [87]:

- gas is supplied to the working cylinder immediately after closing the exhaust valve at the initial stage of the compression stroke [91, 92];
- gas is supplied to the combustion chamber along with pilot fuel at the end of the compression stroke.

The advantages of the first approach are: good mixing of the gas-air mixture during the compression process and the ability to supply gas to the working cylinder under relatively low pressure (Fig. 5.151). Thanks to this last feature, such systems are called *low pressure systems*.

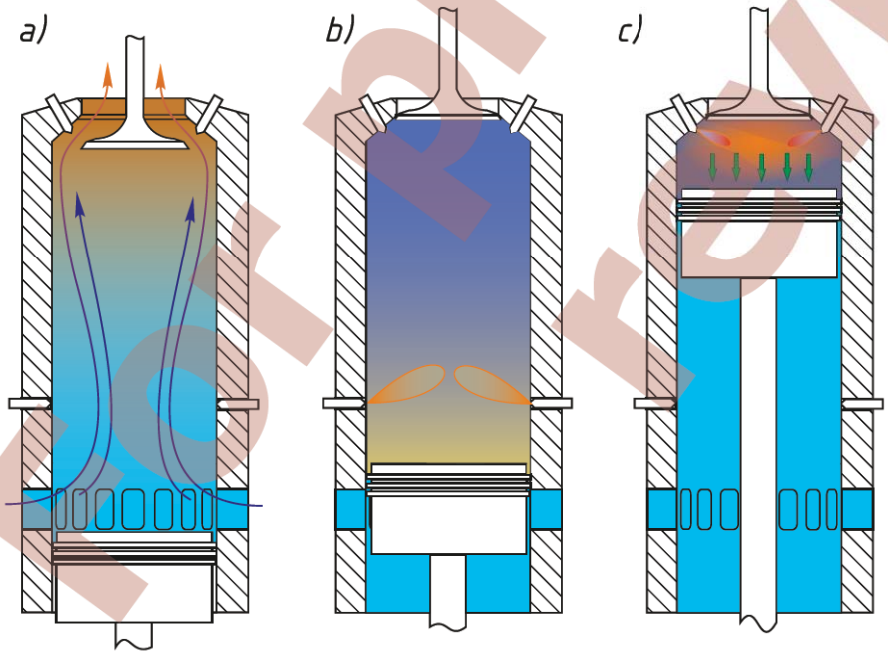


Figure 5.151 – Operating process of a two-stroke gas-diesel engine with gas supplied to the working cylinder immediately after closing the exhaust valve at the beginning of the compression stroke: *a* – purging the cylinder; *b* – gas supply; *c* – ignition, combustion and power stroke (adapted from [9, 83])

The disadvantages of this mixture formation scheme are: the possibility of detonation combustion and the likelihood of leakage of the gas-air mixture through leaks in the piston rings into the sub-piston space during the compression stroke [93, 94].

For preliminary review

Currently, two-stroke low-speed engines manufactured by Winterthur Gas and Diesel Ltd. are equipped with low-pressure systems.

The advantage of the second approach is that, when gas fuel is supplied directly to the combustion chamber, the occurrence of detonation can be completely eliminated. However, gas must be supplied under high pressure, which complicates the fuel system (Fig. 5.152).

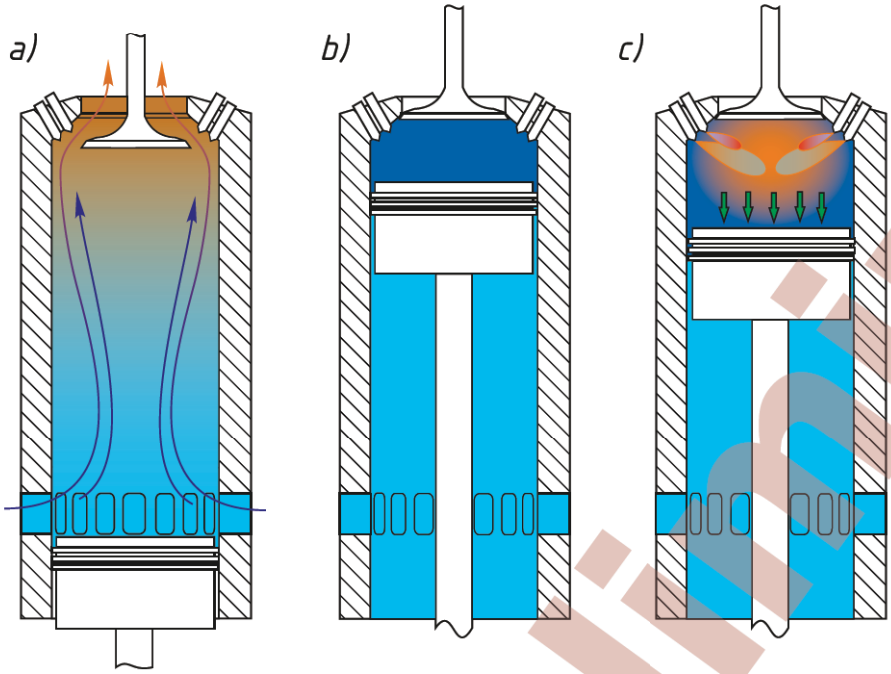


Figure 5.152 – Operating process of a two-stroke gas-diesel engine with gas supply at the end of the compression stroke: *a* – cylinder purging; *b* – gas supply, ignition and combustion; *c* – working stroke (adapted from [9, 83])

Such systems are called *high pressure systems or Direct Injected Gas* (GD) systems.

To compress natural gas, it is necessary to use multi-stage compressors, and to inject liquefied gas, special pumps whose design allows to prevent gas leaks from the working cavity.

On ships transporting liquefied natural gas, the problem of compressing it to high pressure is usually solved in parallel with the need to re-liquefy the cargo being transported.

Currently, low-speed gas-diesel engines from MAN are equipped with high-pressure systems, which are used as main engines, including not only on gas carriers or oil tankers, but also on other types of vessels.

The Japanese company Mitsubishi is developing in the same direction, which, based on diesel engines of the UEC series, is creating its own version of a low-speed gas-diesel engine designated UEC-LSGi.

5.8.3.1 Fuel equipment of two-stroke gas-diesel engines with gas supply on the compression stroke. After an analysis of possible concepts for creating a low-speed gas-diesel engine carried out by Winterthur Gas and Diesel Ltd. (inherited the developments of Wärtsilä) at the beginning of 2011, it was decided to develop the technology of supplying gas fuel to the engine under low pressure. The first studies were carried out at a test base in the Italian city of Trieste. To do this, one cylinder with the diameter of a six-cylinder laboratory diesel engine was converted to run on gas fuel 50 cm. By 2013, this engine was completely converted to run on gas (primary) and liquid (backup) fuels. In this case, using gas fuel, the engine operates on a cycle close to the theoretical Otto cycle.

Since 2013, Wärtsilä has carried out full-scale testing of the RT-flex50DF gas-diesel engine. These developments were a continuation of the work of Sulzer on the creation in the 70s of the last century of low-pressure gas-diesel engines based on diesel engines of the RD and RNMD series. In addition, the company also had experience in creating gas-diesel engines with high-pressure gas supply. So in 1986, Sulzer, together with the Japanese company Diesel United, designed and tested a high pressure gas supply system for the RTA84 engine. At that time, these technologies were not in demand, and only with rising prices for petroleum products and tightening environmental requirements for marine diesel engines, interest in the use of gas fuel in them arose again.

MAN has developed its own version of low-pressure gas-diesel engines. They are issued under

For preliminary review

the symbol GA.

The low-pressure gas supply technology developed for a dual-fuel low-speed engine is designed to burn a lean air-fuel mixture. Previously, this technology was tested by the company on four-stroke medium-speed engines. As shown in Fig. 5.151, gas enters the cylinder after all gas exchange organs are closed, but the pressure still remains relatively low. In practice, gas supply valves are installed at a certain height from the purge ports to provide the necessary time for filling the cylinder with gas until the piston passes the filling holes.

The supply of gas fuel during the compression stroke allows it to be supplied to the engine cylinders at a relatively low pressure of 1.0...1.6 MPa versus 30 MPa when gas is supplied at the end of the compression stroke. This circumstance makes it possible to significantly simplify the fuel system and use cheaper and more reliable screw or centrifugal compressors instead of piston ones used in GI systems. The presence of low pressure in gas lines reduces the risk of gas leaks, and therefore makes them safer.

During the compression process, the gas is well mixed with air and ignited with the help of a pilot portion of fuel. In this case, the size of the pilot fuel portion does not exceed 1% of the cyclic supply of liquid fuel at full load and remains unchanged in all operating modes of the engine on gas fuel The use of natural gas makes it possible to reduce the formation of NO_x by 90% compared to the use of liquid fuel, which is explained by a more uniform temperature distribution throughout the combustion chamber and a smaller number of high-temperature zones in which the formation of NO_x occurs. The use of this technology makes it possible to comply with the International Maritime Organization IMO Tier-III requirements for NO_x emissions without any post-engine exhaust gas treatment.

To ensure dual-fuel operation of the engine, it is equipped with three independent fuel supply systems, each of which can be controlled by an electronic microprocessor module according to a separate program, depending on the fuel used and the engine operating mode.

To supply reserve liquid fuel, a standard accumulator-type fuel system is used, which is typical for all engines of the RT-flex series (clause 5.67). At the same time, the engine retains the ability to operate on heavy fuels over the entire range of load and speed conditions.

The pilot fuel is supplied by a separate low-capacity accumulator system (Fig. 5.153).

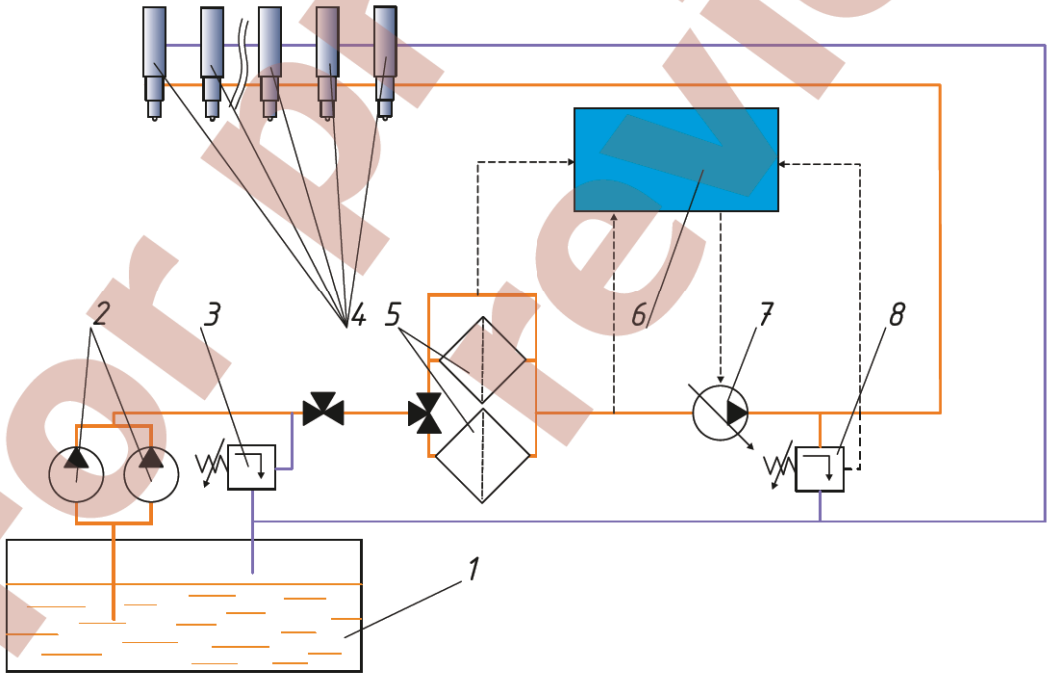


Figure 5.153 – Diagram of the pilot fuel accumulator system of the RT-flex DF gas-diesel engine: 1 – pilot fuel tank; 2 – booster pumps; 3 – bypass valve of the low pressure line; 4 – fuel injectors for pilot fuel injection; 5 – fine filters; 6 – electronic control unit; 7 – high pressure fuel pump; 8 – bypass valve of the high pressure line; — — hydraulic lines; – - – – control lines (adapted from [83])

For preliminary review

This solution makes it possible to obtain a stable supply of small portions of fuel with the possibility of flexible regulation of the operating mode of the pilot ignition system. To supply fuel to the accumulator under high pressure, a radial plunger type fuel pump from L'Orange GmbH (Fig. 5.128) with an electric drive is used. To improve the conditions for ignition of a lean gas-air mixture, two ignition modules are installed on each cylinder, consisting of a swirl chamber with an ignition fuel supply fuel injector installed in it (Fig. 5.154). Light diesel fuel is used as a pilot, but the possibility of using HFO for pilot injection, on which the engine operates as a backup, is being considered.

The internal cavity of the swirl chamber is connected to the combustion chamber using a tangential channel. During the compression process, air flows from the working space into the combustion chamber, swirling and forming a vortex into which, towards the end of the compression stroke, the fuel injector supplies a portion of pilot fuel.

The presence of a vortex promotes good mixing of air with fuel and effective self-ignition. Combustion of fuel leads to an increase in pressure in the swirl chamber, under the influence of which plasma jets through holes in the end of the tangential channel are ejected into the combustion chamber, effectively igniting the lean gas-air mixture.

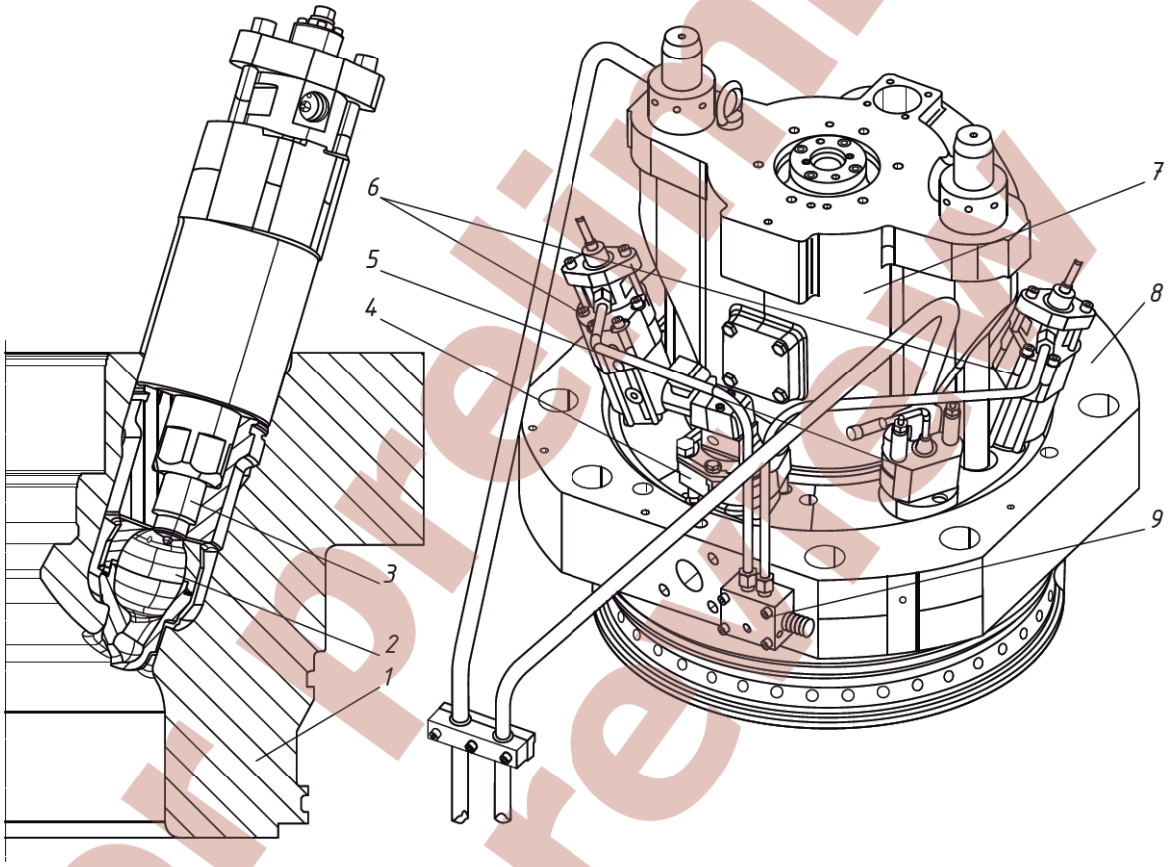


Figure 5.154 – Pilot ignition module for the gas-air mixture of the RT-flex DF gas-diesel engine and its installation on the cylinder cover: 1 – cylinder cover; 2 – vortex chamber; 3 – pilot fuel injector; 4 – air valve; 5 – fuel injector for supplying reserve liquid fuel; 6 – ignition module; 7 – exhaust valve housing; 8 – pilot fuel supply manifold (adapted from [83, 95, 96])

The uniform distribution of plasma jets throughout the volume of the combustion chamber promotes more uniform combustion of the gas-air mixture without local areas of high temperatures characteristic of pilot ignition directly in the engine cylinder. This in turn makes it possible to reduce the level of NO_x in combustion products which is formed mainly in local zones.

The prechamber is made of heat-resistant steel in the form of two separate liners and is cooled along the outer surface with water supplied from the cooling circuit of the cylinder cover. When the engine is running on liquid fuel, to prevent coking of the ignition fuel injectors, they continue to operate in the minimum stable supply mode. The pressure in the accumulator of the pilot fuel injec-

Preliminary review

tion system is maintained at 120 MPa. To prevent fuel from entering the engine components in the event of damage to the high pressure line, all pipelines are placed in a protective shell.

Gas fuel is supplied to the engine working space using two gas valves, which are installed diametrically opposite to each other at a height of approximately 1/3 of the piston stroke. The valves are attached to the cylinder block jacket and gas is supplied to the working cylinder through holes in the sleeve. The general design of the valve is shown in Fig. 5.155.

The valve is opened by a hydraulic piston, and closed and held by a coil spring. The valve drive oil is taken from the exhaust valve drive system, greatly simplifying the system design. The oil flow in the gas valve drive system is controlled using a bistable solenoid valve, which is used in RT-flex engines to control fuel supply (Fig. 5.79).

The signal to the control valve comes from the microprocessor engine control module.

The valve is equipped with a displacement sensor, through which feedback is provided to the control unit. If the valve does not return to the closed state, the control unit automatically stops the gas supply and switches the engine to liquid fuel.

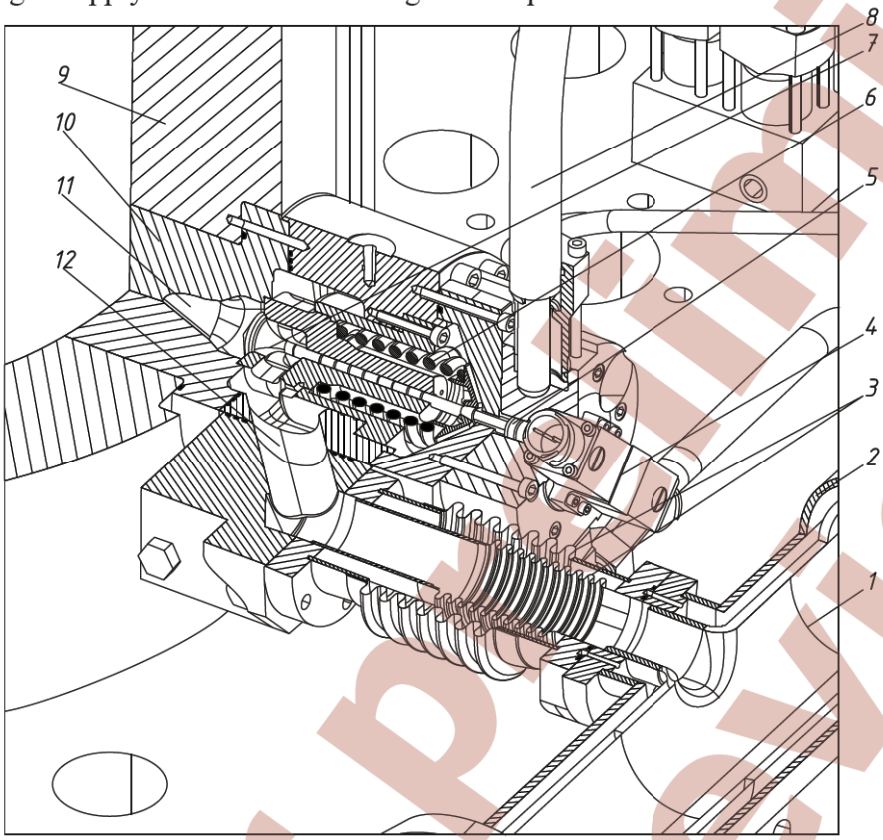


Figure 5.155 – Installation of the gas supply unit on the cylinder liner of the RT-flex 50 DF engine: 1 – gas supply line; 2 – outer shell of the gas supply line; 3 – external and internal bellows; 4 – solenoid control valve; 5 – hydraulic piston of the valve drive; 6 – valve closing spring; 7 – gas valve; 8 – hydraulic line for valve drive; 9 – cylinder liner; 10 – gas valve housing; 11 – channel for supplying gas fuel to the working cylinder; 12 – channel for supplying gas fuel to the gas valve (adapted from [83, 96])

Gas fuel is supplied to the valves through double-walled bellows from gas lines laid on both sides of the engine (Fig. 5.156).

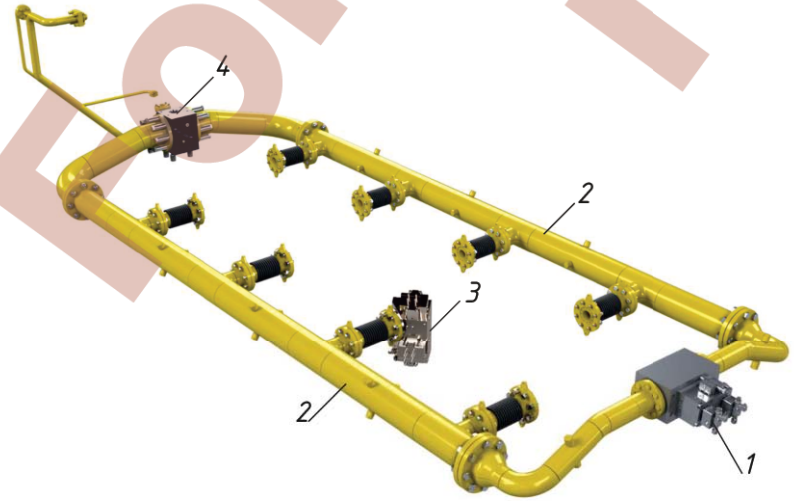


Figure 5.156 – MAN series GA engine: 1 – gas fuel supply control unit; 2 – main pipelines; 3 – module for supplying gas fuel to the cylinders; 4 – nitrogen purge module (adapted from [97])

For preliminary review

All lines are also double-walled, and the space between the walls is used to collect leaked gas. This space is constantly ventilated, and gas sensors are installed at the outlet of the ventilation system, which, if gas content is detected above the established norm, transmits a signal to the control unit to emergency stop the engine or switch it to liquid fuel. For safety reasons, all engine gas pipelines are made of stainless steel.

Operating the engine on gas fuel makes it possible to obtain power at the output flange at the level of 80% of the rated power of the engine on liquid fuel (Fig. 5.157). If it is necessary to obtain higher power, the engine is switched to liquid fuel. For safety reasons, starting and stopping the engine is also carried out using liquid fuel. It should be noted that the resulting power using gas fuel corresponds to the main operating mode of marine engines.

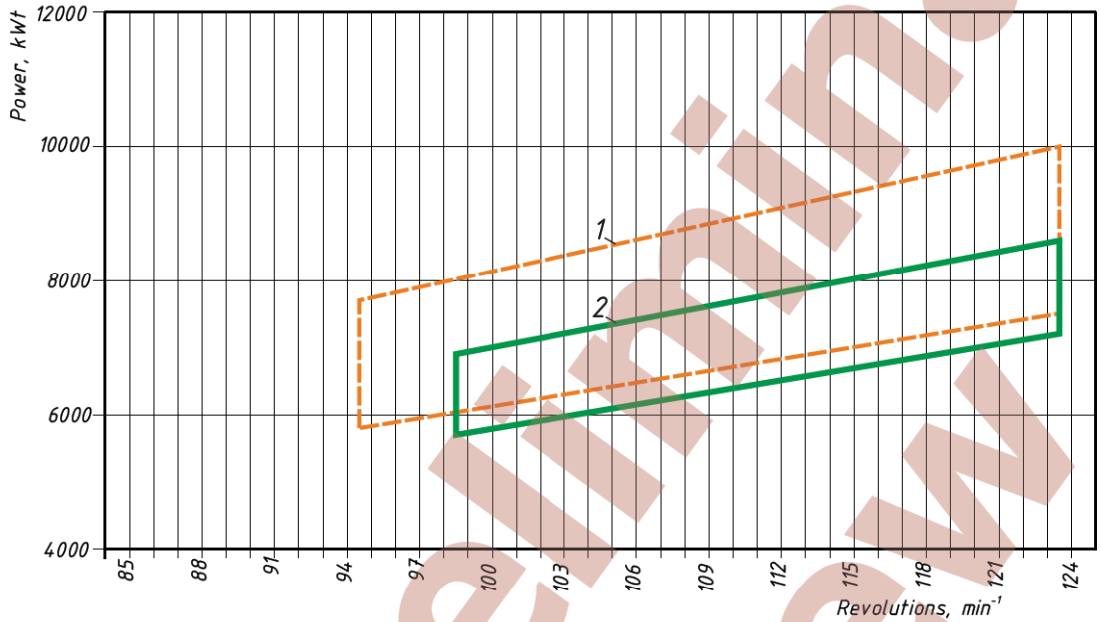


Figure 5.157 – Fields of recommended operating modes for RT-flex50B engines on liquid fuel (1) and RT-flex50DF on gas (2) (adapted from [83, 98])

The main factor limiting power is the occurrence of detonation in the working space of the cylinder. The need to preserve the ability of the engine to operate on liquid fuel required maintaining a sufficiently high degree of compression in it, as a result of which conditions are created in the working cylinder for the occurrence of detonation combustion, which is caused by uncontrolled premature self-ignition of the gas-air mixture. Due to the many sources of self-ignition, wave processes occur in the cylinder, accompanied by high-frequency noise and vibrations (Fig. 5.158).

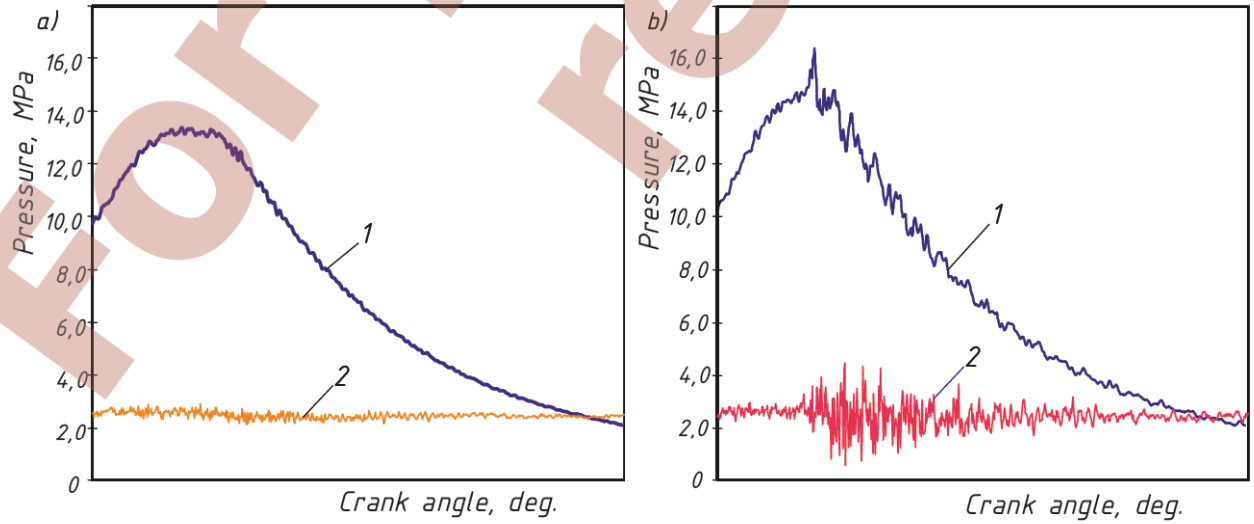


Figure 5.158 – Normal (*a*) and detonation (*b*) combustion of the gas-air mixture in the working cylinder of a low-speed gas-diesel engine RT-flex50DF (adapted from [99])

For preliminary review

At the same time: the cost of work for compressing the mixture in which heat is already being released increases; mechanical loads on engine parts increase; vibration leads to accelerated destruction of engine structural elements. To avoid detonation, it is necessary to maintain the composition of the lean gas-air mixture in a fairly narrow range (Fig. 5.139), which entails a decrease in the energy potential of the charge and, consequently, a decrease in the work it produces.

The resistance of gas fuel to detonation, that is, its ability to resist self-ignition during compression, is determined by the methane number.

The rate of physical and chemical processes leading to detonation depends on the composition of the gas-air mixture. To operate the engine at rated power, the methane number of the gas should not be less than 80; when the power is reduced, the methane number can also be reduced to 80...55 units (Fig. 5.159 *a*). In addition, prolonged compression of the gas-air mixture, accompanied by an increase in its temperature, makes the operating process extremely sensitive to the air temperature at the engine inlet (Fig. 5.159 *b*). The higher the temperature at the beginning of the compression process, the faster the physicochemical processes occur, leading to detonation combustion.

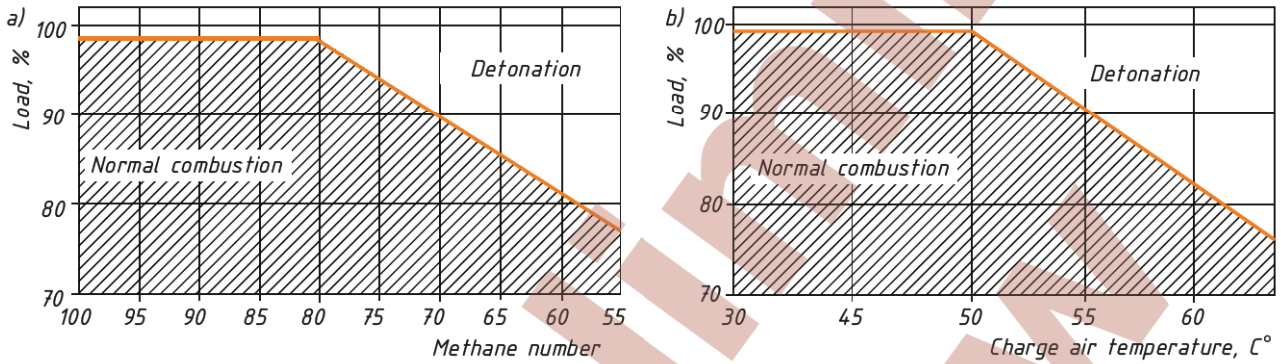


Figure 5.159 – Influence of methane number (*a*) and charge air temperature (*b*) on the occurrence of detonation combustion in the working cylinder of a low-speed engine with gas supply on the compression stroke (adapted from [100])

The need to maintain the composition of the gas-air mixture in a fairly narrow range (Fig. 5.139) does not allow one to quickly increase engine power by significantly increasing the gas supply to the working cylinder, as this can lead to misfires. Reducing the load should also not be accompanied by a sharp decrease in gas supply, as this will lead to detonation combustion. Thus, engines with gas fuel supply on the compression stroke are characterized by a slow response to load changes. The transition from mode to mode should be carried out gradually with delays in intermediate modes to stabilize the composition of the gas-air mixture (Fig. 5.160). This feature makes it difficult to use the engine during maneuvers and requires its conversion to liquid fuel. At the same time, it should be noted that environmental requirements are most stringent in coastal areas, where it is often necessary to maneuver, which to some extent neutralizes the benefits of using gas fuel.

In order to prevent engine operation with detonation, special sensors are installed on each cylinder that record wave processes in the cylinder. If detonation occurs, the sensors transmit a signal to the electronic control unit, which reduces the load on the engine or switches it to liquid fuel. The transition from one type of fuel to another can be carried out without stopping the engine at powers up to 80% of its rated power on liquid fuel. The joint parallel operation of liquid and gaseous fuel supply systems is considered as a promising direction. In this case, part of the energy in the cylinder is obtained by burning gas fuel, which is supplied in the amount necessary to ensure knock-free compression, and the missing part of the energy is obtained by injecting liquid fuel into the combustion chamber.

In addition to knock sensors, the engine is equipped with pressure sensors in the working cylinder, the main task of which is to monitor misfires. If they occur, unburned gas-air mixture may accumulate in the exhaust receiver, which can lead to an explosion and engine damage. Misfires or inefficient fuel combustion can result from malfunctions in the gas-air mixture pilot ignition system.

For preliminary review

In addition, it is possible to control the ingress of gas fuel into the sub-piston space, which is also fraught with consequences. The cause of gas entering the cavity under the piston may be loose closure or damage to the gas supply valves. When the piston passes through the filling holes, gas can enter the sub-piston cavity through leaks in the valves. In addition, some part of the gas-air mixture will constantly seep into the sub-piston cavity through leaks in the piston rings.

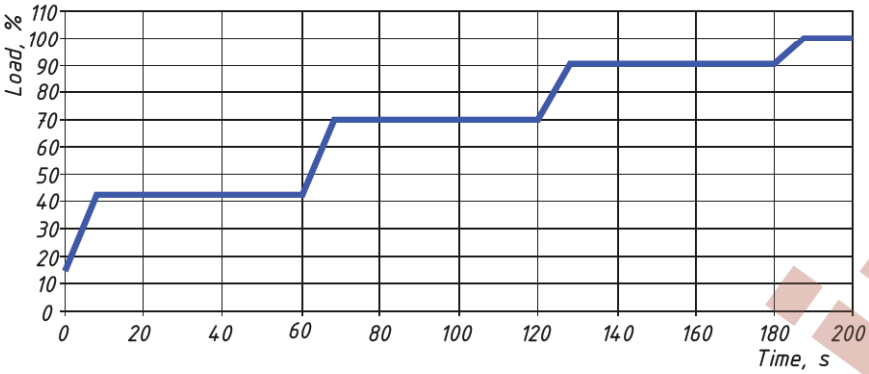


Figure 5.160 – Taking a gas-diesel engine with a low-pressure power supply system to maximum continuous power (adapted from [101])

Due to the above reasons, incomplete combustion of gas fuel leads to the release of a small portion of methane into the exhaust gases. It should be noted that currently there are no current restrictions on methane emissions from marine engines, but it must be borne in mind that the contribution of methane to the greenhouse effect is 20...25 times higher than CO_2. In general, methane emissions into the atmosphere from the ship's power plant do not exceed 0.3%, and the total amount of greenhouse gases emitted is 20...30% lower than that of engines running on liquid fuel.

Currently, Wärtsilä has launched the production of a new generation of low-speed two-stroke engines designed to operate on gas fuel based on previously developed X-series diesel engines. These engines include W-X62DF, W-X72DF, W-X82 and W-X92. In general, the developed technology for converting engines to gas fuel while maintaining the ability to operate on liquid fuel is aimed at both new engines and the modernization of those already in operation. At the same time, according to the company's estimates, the cost of re-equipment will be 20...25% of the original cost of the engine.

5.8.3.2 Fuel equipment of two-stroke gas-diesel engines with gas fuel supply at the end of the compression stroke. Two-stroke low-speed diesel engines of the MC and ME series are used as base models for gas-diesel engines with gas fuel supplied at the end of the compression stroke. Engines modernized for the gas-diesel process received an additional GI index (Gas Injector). In this case, an additional electronic control system for the gas supply process is installed on mechanically controlled (MC) engines, and on electronically controlled (ME) engines, control functions are assigned to the standard control system. Structurally, the gas supply control units for all modifications differ little.

To operate on liquid fuel and to inject the pilot portion, the standard engine fuel system is used. This simplifies the design (there is no special pilot fuel injection system), but does not significantly reduce the consumption of liquid fuel for pilot ignition, the share of which, for this type of engine, is 5...8%.

In part, the issue of significantly reducing fuel costs for pilot ignition for LNG tanker engines is not so pressing, since operating practice shows that at nominal mode the amount of evaporation can only cover 80...90% of the engine's fuel needs. When moving in ballast, the share of evaporation can be 40...50%. Therefore, the fuel system must be able to automatically replace the missing gas fuel with liquid fuel in any ratio. In addition, the calorific value of the gas entering the engine may vary. At the beginning, gas vapors contain a large amount of nitrogen, which, having a lower boiling point (-195.75°C), evaporates first. Therefore, for gas-diesel low-speed two-stroke engines, two modes are considered:

- with a constant supply of pilot fuel (Fig. 5.161 *a*), when at start-up mode and at low loads the engine runs on liquid fuel (MDO, MGO, HFO). Starting from 25% load, a constant ignition supply is established, and the required power is regulated by changing the amount of gas supplied to the

For preliminary review

cylinder;

- when using all available gas (Fig. 5.161 *b*), when at low and medium loads the engine runs on liquid fuel. At higher loads, all gas fuel enters the cylinders, and the required power is adjusted by supplying liquid fuel.

The transition from one type of fuel to another, as well as the transition from mode to mode, is carried out automatically without reducing the power on the shaft over the entire range of possible engine operating loads.

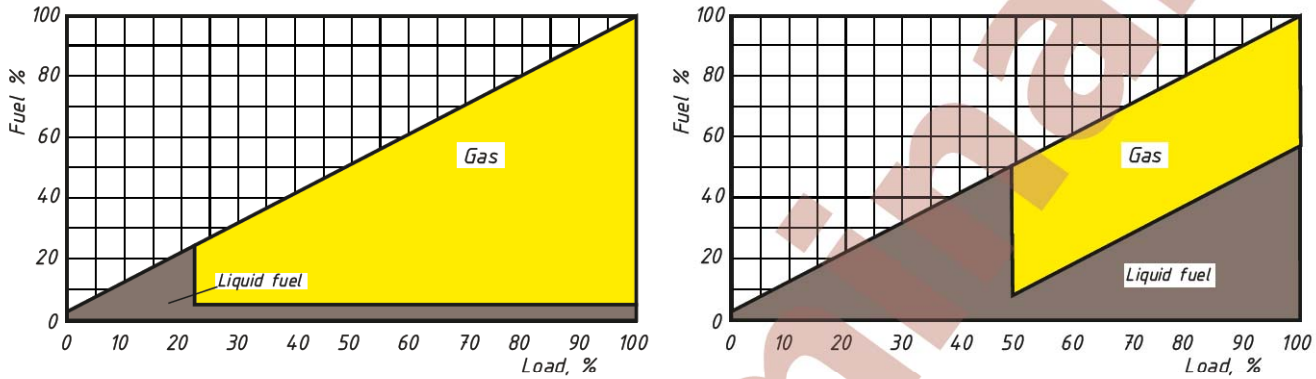


Figure 5.161 – Operating modes of marine low-speed gas-diesel engines: *a* – with a constant amount of ignition fuel; *b* – using all available gas (adapted from [102])

Equipment for supplying gas fuel under high pressure includes compressors, heat exchangers, a system for supplying gas fuel to the working cylinders, gas supply control modules and gas injectors.

On gas carriers carrying LNG, Cryostar compressors are widely used to recompress the evaporating cargo and to supply gas to the engine at high pressure.

The general design of the 6LP250-5S-1 series piston compressor is shown in Fig. 5.162.

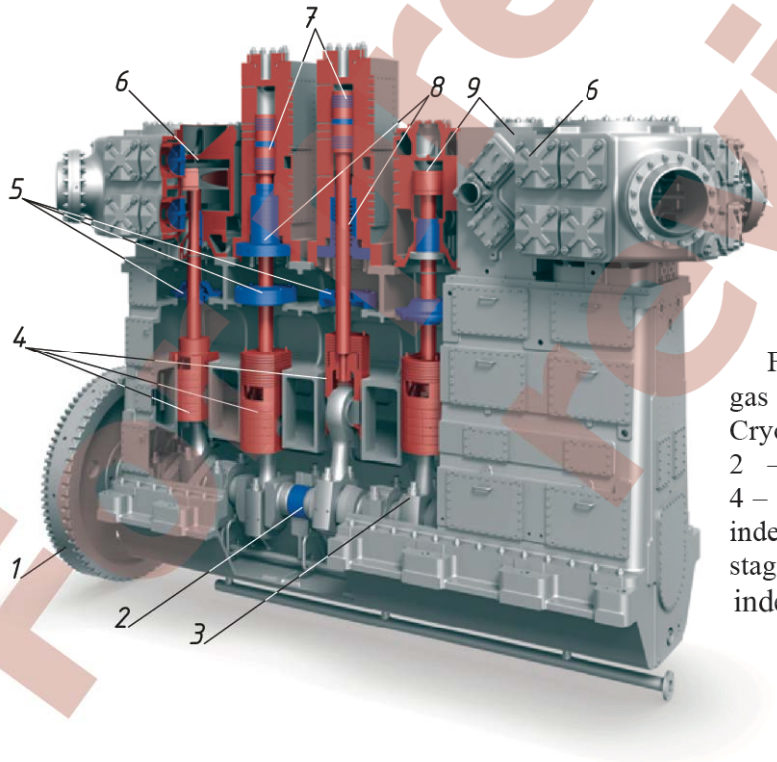


Figure 5.162 – General design of the gas compressor series 6LP250-5S-1 from Cryostar: 1 – flywheel and drive flange; 2 – crankshaft; 3 – connecting rod; 4 – slider; 5 – rod seal; 6 – first stage cylinders; 7 – cylinders of the fourth and fifth stages; 8 – rod seals; 9 – second stage cylinders (adapted from [100])

The compressor with a vertical cylinder arrangement has five stages of compression and allows the output to produce gas at a pressure of up to 30 MPa. In this case, the fourth and fifth stages are combined in one working cylinder. In total, the compressor has two working cylinders for each

Preliminary review

compression stage. The main feature of this compressor is the ability to operate in a wide range of suction gas temperatures from -160 to -40°C. This is achieved by careful selection of materials and a number of original design solutions. Considering that the amount of evaporation from tanks can vary widely, two compressors are installed on the vessel, each of which is able to fully meet the engine's needs. In addition, each compressor has the ability to reduce its capacity by half by turning off half of the working cylinders using bypass valves actuated by compressed nitrogen.

The greatest peak in gas emissions occurs during cargo operations, when the main engine does not consume fuel at all, at which point the compressors operate for liquefaction. As the ship moves, some of the gases are diverted to power the main engine, and unused fuel is re-liquefied and returned to the cargo tanks.

On ships not intended for the transportation of gases, the power plants of which are converted to gas fuel transported on board in special containers, in-line gas compressors with horizontal cylinders are used. A cross section of such a compressor from Cryostar is shown in Fig. 5.163.

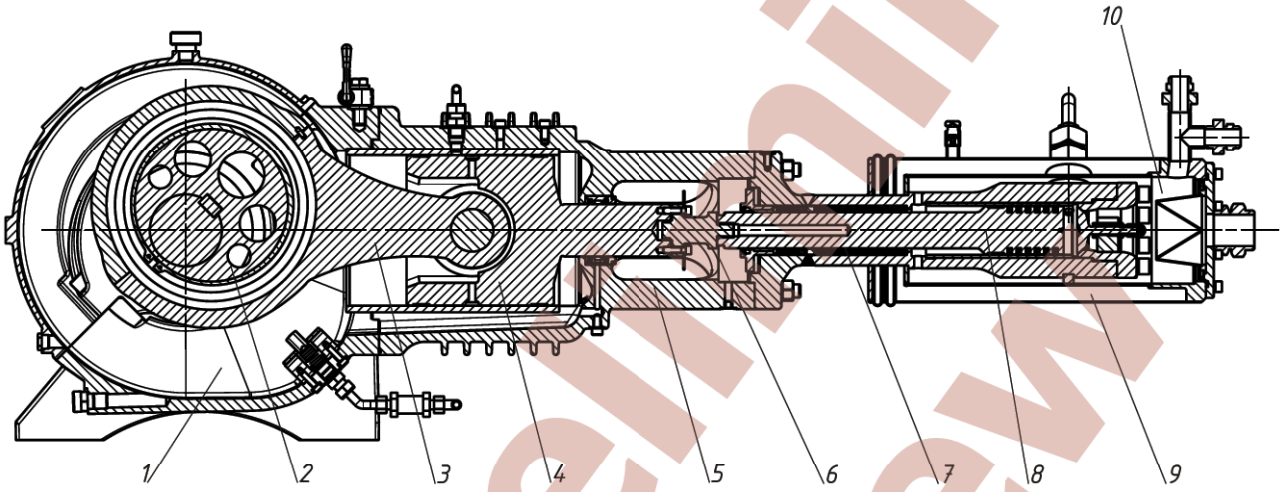


Figure 5.163 – High-pressure gas compressor of the HPP3 60/110 series from Cryostar: 1 – crank chamber; 2 – crank; 3 – connecting rod; 4 – crosshead; 5 – sealing chamber; 6 – drive rod of the working piston; 7 – working cylinder with a sealing system; 8 – working piston; 9 – vacuum heat-insulating chamber; 10 – receiving chamber (adapted from [100])

These compressors allow you to obtain an outlet pressure from 15 to 30 MPa. The amount of fuel supply pressure depends on the engine operating mode. Compressors of this type are highly efficient. This is mainly achieved due to low mechanical losses and good thermal insulation of the working cylinder, including a vacuum heat-insulating chamber.

After cooling the gas in the heat exchanger, it enters the gas supply line of the engine, which simultaneously functions as a pressure accumulator.

All gas pipelines on the engine are all-welded and only at the connection points of the tubes that discharge gas fuel to the supply control units are flanged connections necessary for servicing the elements of the gas system. The piping is designed to compensate for thermal expansion as the engine heats up. All pipes of the gas system are designed for a pressure 50% higher than the working pressure, and during factory tests they are pressurized with a pressure 150% higher than the working one. All gas pipes are placed in protective shells that can withstand the pressure that may occur if the main line ruptures. The internal space between the shell and the pipeline is connected to a forced ventilation system, which provides approximately 30 air changes within an hour. Ventilated cavities also include cavities adjacent to the main elements of the fuel system, in which gas leakage may occur. At the outlet of the ventilation system, a gas presence sensor is installed, connected to the alarm system. The location of the pipelines is selected so that they are not damaged in the event of heavy objects falling during engine repair and maintenance. Otherwise, the safety measures are the same as in previously discussed installations with four-stroke gas-diesel engines. A fragment of the gas main is shown in Fig. 5.164.

For preliminary review

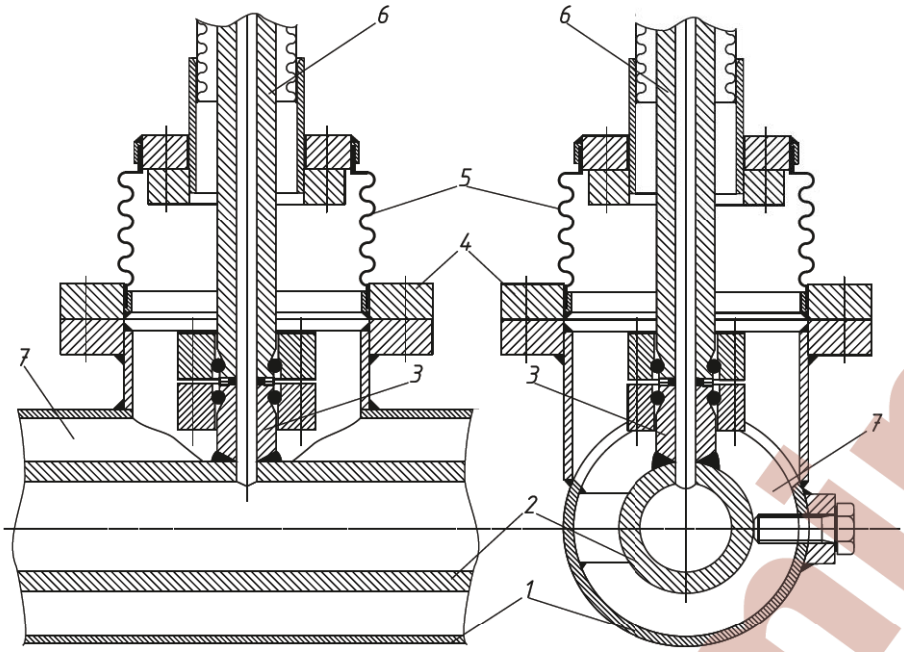


Figure 5.164 – Fragment of the gas line of a gas-diesel engine of the ME-GI series: 1 – protective shell; 2 – gas main; 3 – connecting fitting; 4 – connecting flange; 5 – protective corrugated shell; 6 – gas outlet pipes to the supply control unit; 7 – ventilated space (adapted from [103])

To increase the safety of engine operation, the power plant includes a system of inert gases, which makes it possible to purge both the entire gas fuel supply system and its individual elements under a pressure of 0.4...0.8 MPa. Such cleaning is a mandatory procedure when switching to a diesel cycle or if any part of the gas supply system is damaged.

As already noted, gas-diesel engines are retrofitted with a system for supplying gas fuel to the working cylinder. The power supply system for a gas-diesel engine is shown schematically in Fig. 5.165 *a*.

From Figure 5.165 *b* it is clear that gas is supplied to the combustion chamber immediately after the pilot portion of liquid fuel is supplied to the cylinder and ignited. In this way, a high degree of fuel burnout is achieved and the danger of gas entering the sub-piston space through leaks in the piston rings is prevented.

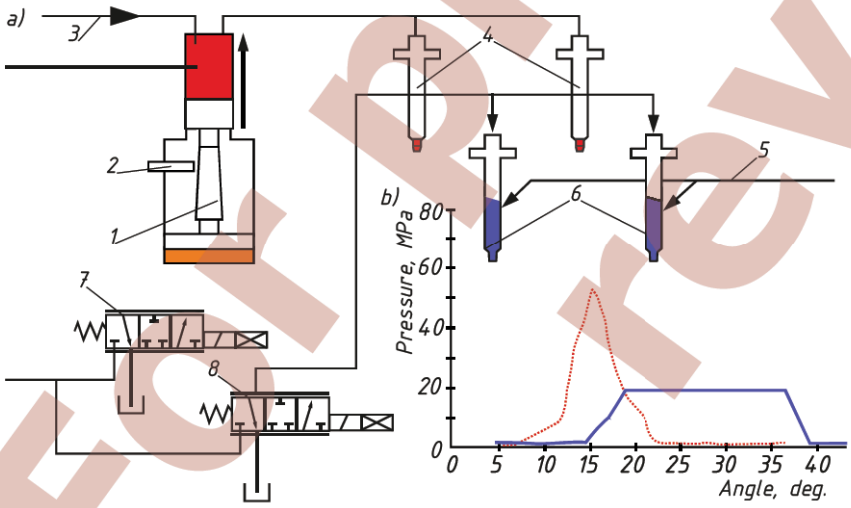


Figure 5.165 – Diagram of the fuel system of a gas-diesel engine of the ME-GI series (*a*) and the laws of supply of pilot (- - - -) and gas fuel (——) (*b*): 1 – injection pump with hydraulic drive; 2 – liquid fuel supply quantity sensor; 3 – liquid fuel supply; 4 – fuel injectors for liquid fuel injection; 5 – gas supply; 6 – gas injectors; 7 – spool valve for fuel injection pump control (FIVA); 8 – spool valve for controlling the hydraulic drive of injectors (ELGI); 9 – supply of control oil under pressure 30 MPa (adapted from [104])

All gas supply control elements are assembled in one module, which includes: a gas accumulator, a main shut-off valve with a hydraulic drive, a valve for purging the system with inert gas, and a valve for controlling the hydraulic drive of the injectors.

The module itself is attached to the cylinder cover, which has internal drillings for supplying gas from the control module to gas injectors installed in the cylinder cover next to the liquid fuel injection fuel injectors.

The general structure of the gas supply control module is shown in Fig. 5.166. The module dia-

For preliminary review

gram and the order of its operation are presented in Fig. 5.167.

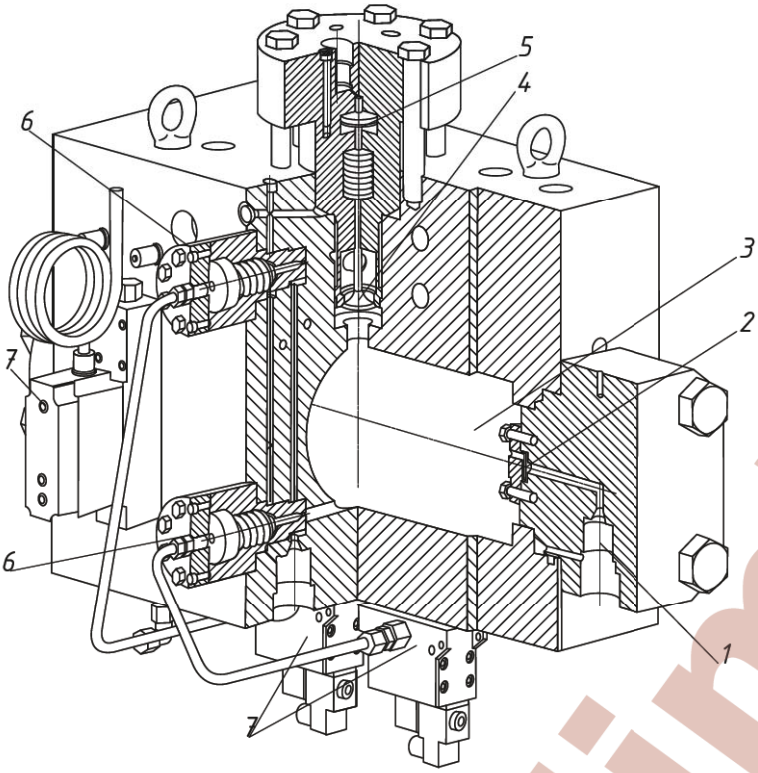


Figure 5.166 – Gas supply control module: 1 – gas fuel input unit; 2 – check valve; 3 – pressure accumulator cavity; 4 – main shut-off valve; 5 – piston of the main shut-off valve drive; 6 – valve for purging the system with inert gas; 7 – control unit for the main shut-off valve; 8 – control unit for the valve for purging the system with inert gas (adapted from [105])

Gas fuel from the supply line flows through a check valve into a pressure accumulator, made as a cavity in the control module housing. The accumulator capacity is equivalent to approximately 20 gas cycles at rated load.

The presence of an accumulator in the control unit serves to reduce the pressure drop during fuel injection. Stable pressure is necessary so that the control system can correctly determine the injection time, which sets the cyclic flow rate.

In the absence of a control signal to the control unit, the main shut-off valve is closed and gas does not flow to the gas injectors (Fig. 5.167 *a*). When an electrical signal is received from the engine control unit to the main shut-off valve control unit, its spool moves and supplies control oil to the main shut-off valve hydraulic drive mechanism. The valve opens and gas flows to the fuel injectors, the needle valves of which remain closed at this moment (Fig. 5.167 *b*). Filling the channels between the module and the injectors, the gas acts on the pressure sensor. Information about the actual pressure received from the sensor is sent to the engine control unit and is used to calculate the required injector opening time to ensure the specified cyclic flow rate.

Based on the information received, the control unit generates a signal supplied to the injector hydraulic drive control unit. Under the influence of the signal, the block spool moves and supplies control oil to the injector drive (Fig. 5. 167 *c*). By opening, the injectors supply gas to the combustion chamber of the engine.

After removing the control signal from the injector hydraulic drive control unit, the spool, moving, switches the oil from the hydraulic drive circuit to drain, as a result of which the injector valves close and the fuel supply stops (Fig. 5.167 *d*).

Removing the signal from the control unit of the main shut-off valve leads to its closure, and the system returns to its original state (Fig. 5.167 *a*).

In the considered scheme for supplying gas to the working cylinder, a two-stage connection is used – first through the main shut-off valve, and then through the injector valves. This solution allows you to increase the level of operational safety, and, if necessary, clean damaged cavities by blowing them through special valves with inert gases.

The general structure of a gas injector is shown in Fig. 5.168.

Gas injectors are installed, two per cylinder, in special wells made in the cylinder cover and located in close proximity to the liquid fuel injectors. Gas is supplied to the injector through drillings

For preliminary review

in the cylinder cover and enters the shut-off cone of the needle valve through holes in its housing. To prevent gas leaks, O-rings are installed between the injector housing and the cylinder cover. The cavities located outside the seals are connected to the general ventilation system.

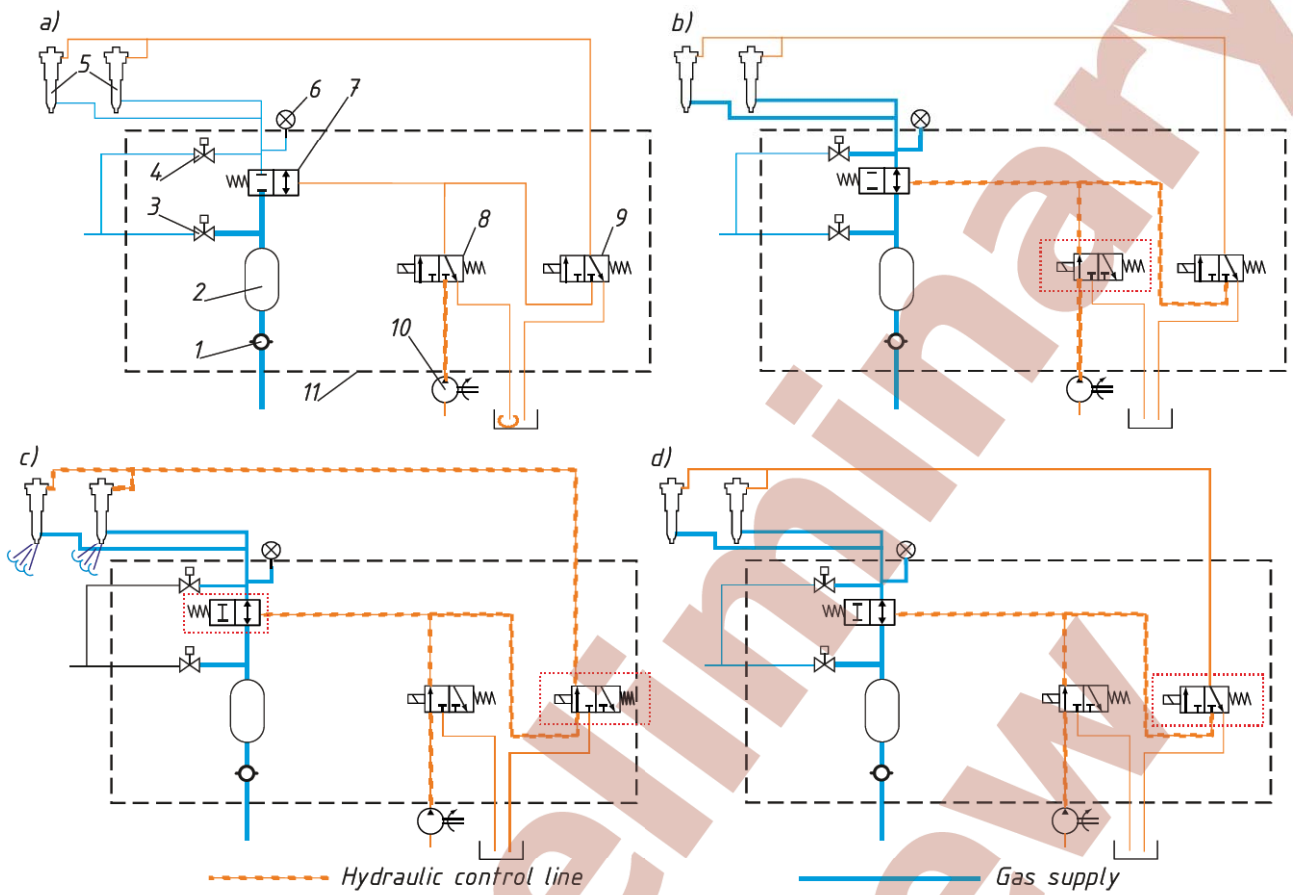


Figure 5.167 – Diagram of the gas supply control module and its operating procedure: *a* – there is no gas supply to the gas injectors; *b* – gas supply to gas injectors; *c* – supply of control oil to the injectors, fuel injection; *d* – closing of the injector valves, stopping injection. 1 – gas line; 2 – accumulator; 3 – purge valve; 4 – emergency release valve; 5 – gas injectors; 6 – gas pressure sensor; 7 – main shut-off valve; 8 – control unit for the main shut-off valve; 9 – control unit for hydraulic drive of injectors; 10 – control oil pump; 11 – supply control module (adapted from [104])

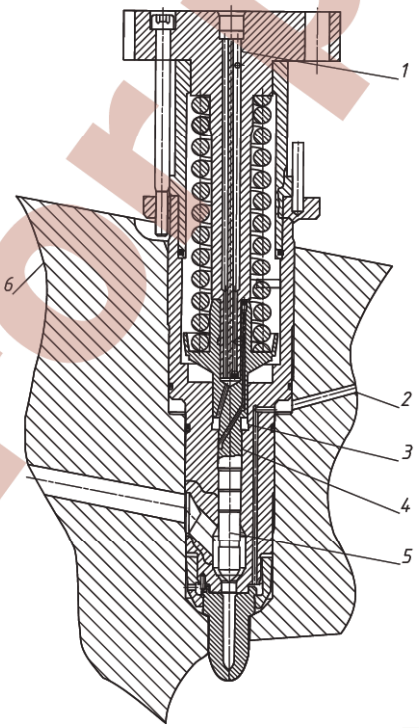


Figure 5.168 – Gas injector of ME-GI series engines: 1 – control oil supply channel; 2 – ventilation duct; 3 – cavity for hydraulic control of the needle valve; 4 – needle valve; 5 – gas cavity; 6 – cylinder cover (adapted from [106])

For preliminary review

In the closed state, the needle valve is held due to the force of the spring acting on the plate, made as one piece with it. At the bottom of the needle valve there is a collar, precision fitted to the housing, which acts as a hydraulic piston. Through channels in the guide and in the housing of the needle valve, oil from the supply control module enters the annular cavity under the collar, forcing the needle valve to open. To prevent gas leakage between the needle valve and the fuel injector housing, a special sealing oil is constantly supplied into the gap between them at a pressure 0.2...0.3 MPa higher than the gas pressure in front of the fuel injector. The sealing oil flows through a system of channels into the annular groove on the housing of the needle valve.

5.8.3.3 Comparison of low and high pressure systems of gas-diesel low-speed marine diesel engines.

As shown above, today there are two fundamentally different approaches to organizing the operation of low-speed gas fuel engines, each of which has its own advantages and disadvantages.

The advantages of low pressure systems include:

- good mixing of the gas-air mixture during the compression process;
- use of a minimum portion of pilot fuel to ignite the gas-air mixture;
- gas supply to the working cylinder under relatively low pressure, which reduces the likelihood of leaks, simplifies the fuel system, and increases the safety of its use;
- cheaper and more reliable screw or centrifugal compressors can be used to supply gas fuel;
- the ability to meet IMO Tier-III requirements for NO_x emissions without any subsequent treatment of exhaust gases after the engine;
- significantly lower content of sulfur oxides SO_x and especially particulate matter (soot) in the exhaust gases compared to engines equipped with high-pressure systems (Fig. 5.169).

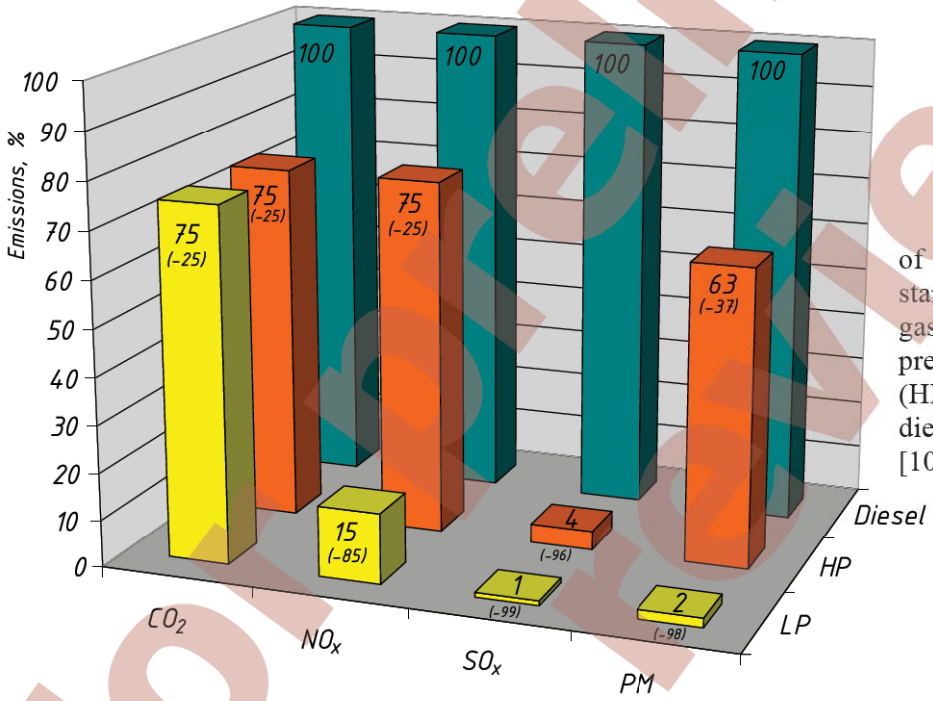


Figure 5.169 – Comparison of emissions of harmful substances from exhaust gases of gas-diesel engines with low-pressure (LP) and high-pressure (HP) systems in relation to a diesel engine (adapted from [107])

Disadvantages of low pressure systems:

- the possibility of detonation and, as a consequence, higher requirements for the quality of gas fuel;
- inevitable leakage of the gas-air mixture through the piston rings into the sub-piston space during the compression stroke, as well as the likelihood of gas entering the sub-piston space in the event of damage to the gas valve;
- the presence of methane in the exhaust gases in a sufficiently large amount, the contribution of which to the greenhouse effect is 20...25 times higher than CO_2, increases the equivalent greenhouse gas emissions from engines, thereby reducing the effect of using gas fuel by 10...15% (Fig. 5.170);

Preliminary review

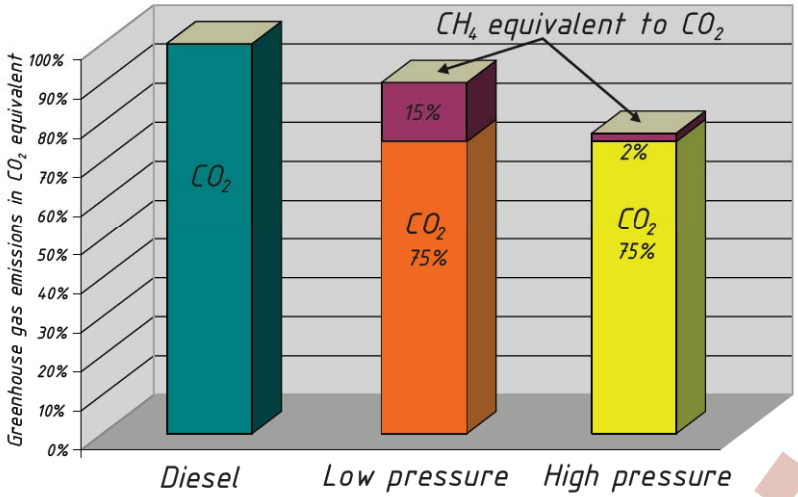


Figure 5.170 – Greenhouse gas emissions (CO_2+CH_4 in CO_2 equivalent) from exhaust gases of gas-diesel engines with low-pressure (LP) and high-pressure (HP) systems in relation to a diesel engine (adapted from [94])

- engine power limitation at 80% of the nominal when operating on gas fuel;
- slow response to load changes;
- higher specific gas fuel consumption compared to high-pressure systems, especially when the load decreases (Fig. 5.171).

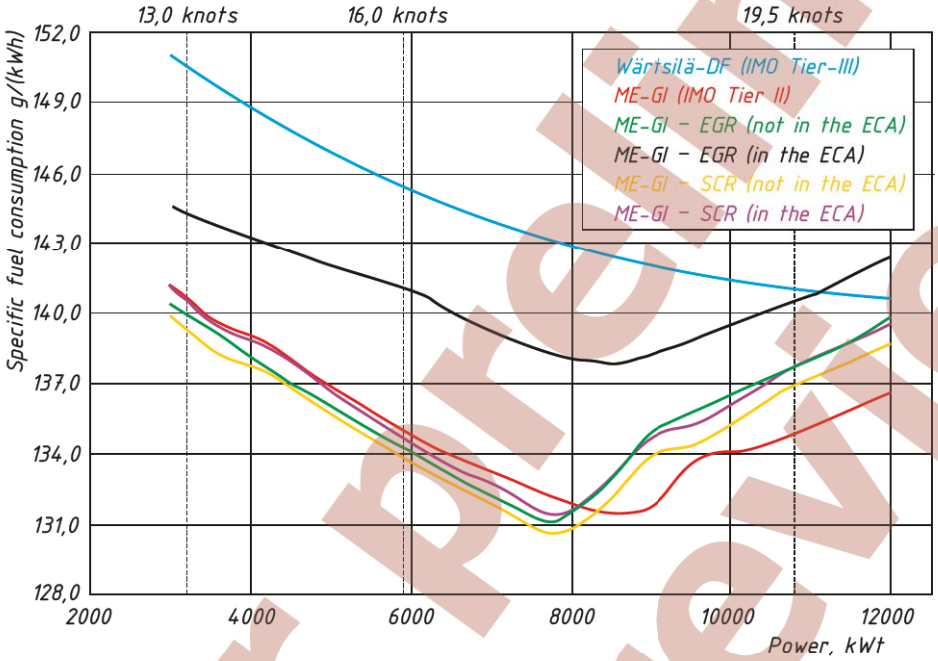


Figure 5.171 – Change in the corrected specific gas fuel consumption for main engines with low (Wärtsilä DF) and high pressure (ME-GI) systems, taking into account the ignition fuel corrected to gas fuel. EGR – Exhaust Gas Recirculation; SCR – Selective Catalytic reduction; ECA – Emission Control Area (adapted from [108])

Advantages of direct injection systems:

- when gas fuel is supplied directly into the combustion chamber, the occurrence of detonation can be completely eliminated, therefore, there are no restrictions on engine power when operating on gas fuel, and the requirements for the quality of gas fuel are less stringent;
- gas fuel cannot enter the sub-piston space;
- lower greenhouse gas emissions compared to low-pressure systems(Fig. 5.170);
- lower specific gas fuel consumption compared to low-pressure systems, especially at partial loads (Fig. 5.171).

Disadvantages of direct injection systems:

- the use of high pressure gas complicates the fuel system and increases the requirements for its safety;
- high consumption of liquid fuel for pilot ignition of the gas-air mixture;
- to compress natural gas, it is necessary to use multi-stage compressors, which reduces the eco-

For preliminary review

nomic performance of the power plant due to the increasing costs of compressing gas fuel to high pressure;

- inability to meet IMO Tier-III requirements for NO_x emissions without the use of additional methods to reduce exhaust gas toxicity, such as: exhaust gas recirculation EGR; exhaust gas treatment in a catalytic reactor SCR, etc., especially for special emission control zones. The use of additional methods leads to an increase in specific fuel consumption, however, even in this case it remains lower than that of engines with low pressure systems (Fig. 5.171).

As an important criterion when choosing a particular type of gas-diesel engine, it is necessary to take into account its cost. Thus, according to Wärtsilä estimates, equipping a base diesel engine with a low-pressure gas fuel supply system increases its cost by 20...30%, and with a high-pressure system by 35...40%.

For greater clarity, the most significant design differences and features in the organization of the operating process of gas-diesel engines with low and high pressure systemsare presented in Table 5.8.

Table 5.8 – Comparison of features of gas-diesel low-speed engines with low and high pressure systems

Features of design and organization operating processe	Low pressure	High pressure
Gas fuel power	80% of N_e^{lf}	100% of N_e^{lf}
Sensitivity to gas quality (methane number)	not lower than 80	No
Sensitivity to inlet air temperature	There is	No
Pilot fuel fraction	≈ 1%	3…8%
Availability of a separate pilot injection system	Eat	No
Possibility of using HFO as an igniter	In perspective	Yes
Possibility of using HFO as a backup	Yes	Yes
Ability to operate on two fuels simultaneously (except pilot)	No / In the future	Yes
Mixing formation	With stirring	No stirring
Gas fuel supply pressure	1.6 MPa	30.0 MPa
Maximum cycle pressure	Increasing	Without changes
The occurrence of detonation when the load changes	Maybe	No
Possibility of misfire	Eat	No
Possibility of explosion in the piston space	Eat	No
Possibility of explosion in the exhaust receiver	Eat	No
Possibility of gas leaks into the engine room	Low	High
Efficient efficiency on gas fuel	Without changes	Increasing
Efficient efficiency on liquid fuels	Decreasing	Without changes
Compliance with IMO NO_x emission standards	Tier III	Tier II
CO_2 emissions from exhaust gases g/(kWh)	485.7	452.4
Methane emissions from exhaust gases g /(kWh)	3…6	≈ 0.5
GWP (global warming potential in CO_2 equivalent) from liquid fuel (^{lf})	Below 10%	Below 23%
Ability to switch from one fuel to another under load	Up to 80% of N_e^{lf}	No limits
Engine response to load changes	Slow motion	Without changes

It is obvious that both approaches to organizing the operating process in gas-diesel engines have their advantages, which for a certain type of installation can be decisive, and the development of designs will make it possible to get rid of most of the disadvantages inherent today.

In this regard, it is obvious that in the foreseeable future both directions will develop in parallel. Despite significant differences in approaches, the availability of alternative solutions allows the customer to choose a power plant that best suits his requirements.

For preliminary review

5.8.3.4 Fuel system of engines running on liquefied petroleum gas and other alternative fuels. The gas fuel system of engines running on liquefied petroleum gas is distinguished by the fact that gas fuel enters the power system in a liquefied state and all elements of the system are adapted to work with liquid rather than gas. This circumstance determines a number of design features that will be discussed below. The general diagram of the gas system of a gas-diesel engine running on liquefied gas is shown in Fig. 5.172.

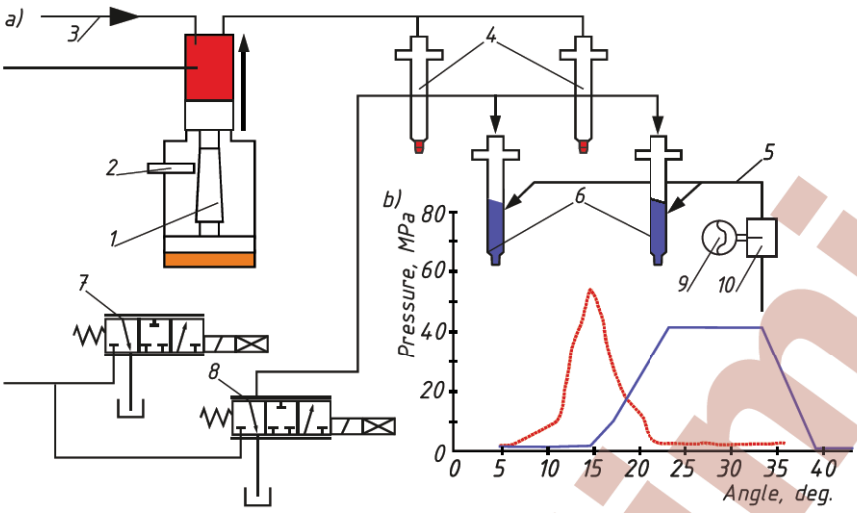


Figure 5.172 – Diagram of the fuel system of a gas-diesel engine of the ME-GI series, operating on liquefied gas (*a*) and the laws of supply of ignition and gas fuel (*b*): 1 – injection pump with hydraulic drive; 2 – liquid fuel supply quantity sensor; 3 – liquid fuel supply; 4 – fuel injectors for liquid fuel injection; 5 – gas supply; 6 – gas injectors; 7 – spool valve for fuel injection pump control (FIVA); 8 – spool valve for controlling the hydraulic drive of injectors (ELGI); 9 – gas accumulator of diaphragm type; 10 – liquefied gas supply control unit; - - - - – pilot fuel supply; ——— – gas supply (adapted from [104])

P G supply system differs from the design of the LNG supply control module in that liquefied gas is supplied to the system not by a compressor, but by a plunger pump at a pressure of 50...55 MPa. In addition, the liquefied gas control module uses a gas accumulator with a diaphragm seal to stabilize the pressure. The design of the injection control module is shown in Fig. 5.173.

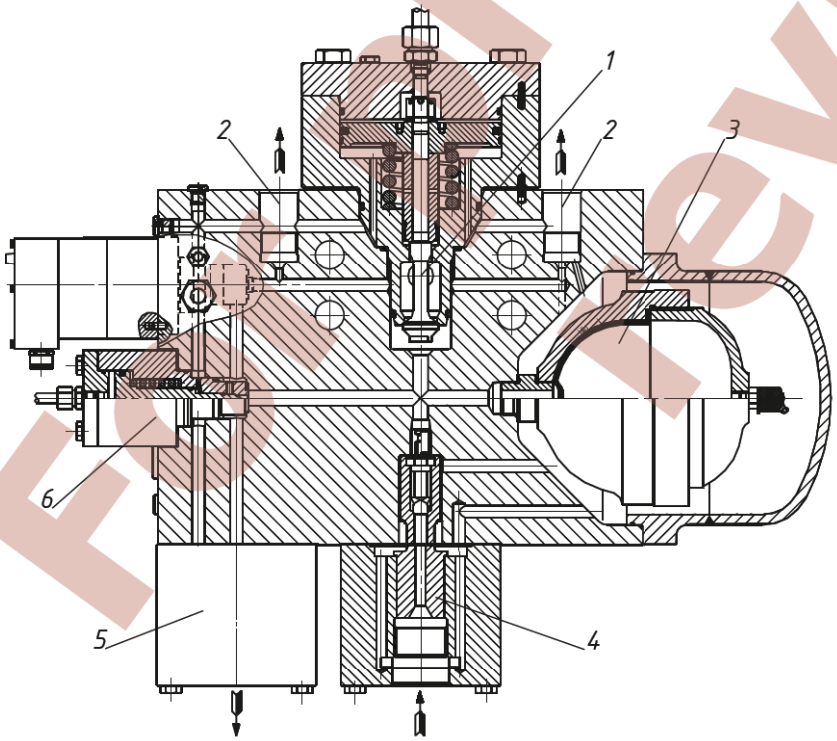


Figure 5.173 – Liquefied gas supply control module: 1 – main shut-off valve; 2 – discharge of liquefied gas to gas injectors; 3 – gas accumulator with a separating diaphragm; 4 – supply of liquefied gas under pressure 50…55 MPa; 5 – device for connecting to the ventilation system; 6 – valve for purging the system with inert gas (adapted from [109])

Unlike the gas fuel supply control algorithm, when injecting liquefied gas, the main shut-off valve does not close with each injection, but serves to disconnect the high pressure line of the injec-

For preliminary review

tors when the engine is not running. The valve is driven by compressed air from the ship's general system.

To clean the system from residual gas fuel, the unit is equipped with a valve for purging the module cavities with inert gas, which uses nitrogen under a pressure of 0.3...0.4 MPa.

The fuel injector for liquefied gas injection is shown in Fig. 5.174.

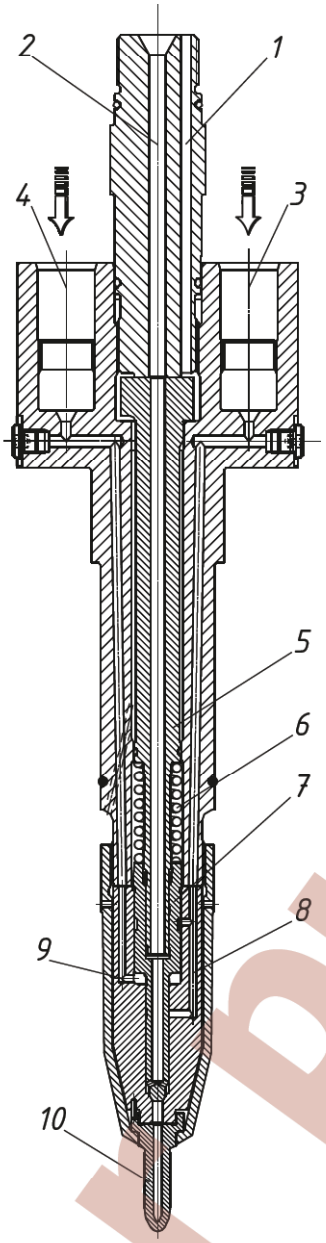


Figure 5.174 – Fuel injector for liquefied gas injection: 1 – channel for monitoring gas leaks; 2 – channel for supplying liquefied gas to the fuel injector; 3 – sealing oil supply channel; 4 – control oil supply channel; 5 – needle valve guide; 6 – spring; 7 – needle valve housing; 8 – channel for supplying sealing oil to the needle valve; 9 – channel for supplying control oil to the cavity of the hydraulic drive of the needle valve; 10 – nozzle tip (adapted from [109])

The injectors are connected to the control unit by high-pressure pipes. Liquefied gas is supplied to the needle valve of the fuel injector through the central channel in the needle valve guide and then through the channel in the housing of the valve itself. Otherwise, the design and principle of operation of the fuel injector are similar to the previously discussed design of the fuel injector for gas fuel.

Tests carried out by MAN have shown that liquefied petroleum gases can spontaneously ignite in the working cylinder even without igniting liquid fuel. At rated power modes, the nature of the operating process in a purely gas cycle is not much different from work in a gas-diesel cycle. When switching to partial loads, a significant increase in the rigidity of the operating process is observed.

In this regard, it was decided to retain pilot ignition in liquefied gas engines over the entire range of operating modes.

Tests in laboratory conditions have shown that fuel equipment designed for the use of liquefied gas can in the future be used to operate methanol engines.

For preliminary review

5.9 Mixture formation in diesel engines. Performance indicators of the fuel equipment of modern diesel engines and their impact on the operating process

The mechanical and thermal stress of the parts of the cylinder-piston group, the crank mechanism, the distribution of power between the cylinders, and ultimately the efficiency, reliability and service life of the diesel engine largely depend on the quality of the fuel system. The dynamics of heat release in the cylinder directly depends on the perfection of the design of the fuel equipment, which significantly determines the nature of the engine's operating process, its economic and environmental characteristics.

The main factors influencing the process of heat release in the engine are the chosen method of mixture formation, the characteristics of the process of supplying fuel to the combustion chamber and the quality of mixture formation.

Factors influencing the quality of mixture formation and combustion of fuel can be divided into independent and dependent on the performance of fuel equipment. The first group of factors includes: the quality of the engine air supply, the chosen method of mixture formation, the degree of mobility of the air charge and its distribution relative to the jets of injected fuel (determined by the shape of the combustion chamber). The second group of factors that have the greatest impact on the operating process of a diesel engine include: the quality of fuel atomization, injection start angle, injection characteristics and duration.

Next, we will consider in more detail the second group of factors as being directly related to the topic of this chapter. Issues related to the first group of factors can be studied in specialized literature.

5.9.1 Fuel injection and atomization processes

In the power supply systems of modern diesel engines, liquid fuel supplied to the nozzle under pressure is the main source of energy needed to produce an aerosol.

Spraying is the result of internal disturbances in the flow, as well as the interaction of the liquid jet with the walls of the nozzle channel and the surrounding gaseous environment. The complex action of these factors leads to the destruction of the flow into individual droplets. At the same time, the contact surface of the fuel with the heated air charge increases significantly, which accelerates the processes of heating, evaporation, mixture formation and combustion.

For liquid injectors, atomization quality depends on the design of the nozzle, the properties of the fuel and the injection pressure. In addition, in diesel engines, fuel is injected into a combustion chamber of limited volume, in which the air charge has a fairly high temperature and is under a pressure of 4...16 MPa. These are additional factors influencing the quality of liquid fuel atomization.

The atomization process occurs in several stages and is characterized by a number of parameters, which can be divided into macroscopic and microscopic. Macroscopic parameters include parameters that characterize the geometric dimensions and shape of aerosol formations formed during the process of fuel atomization and are called *atomization plume*. Microscopic parameters include parameters characterizing the size of aerosol droplets, their homogeneity, and the distribution of aerosol droplets in the plume by size and speed.

5.9.2 Stages and macroscopic parameters of the injection process, their influence on mixture formation in diesel engines

In Fig. 5.175 shows eight frames of high-speed filming, which show the development of a atomization plume of light diesel fuel, filmed within one injection. The injection conditions are as close as possible to the conditions characteristic of a high-speed engine. In the case under consideration, fuel atomization was carried out in an experimental chamber filled with nitrogen through a fuel injector hole with a diameter 0.4 mm at a pressure in front of the nozzle of 70 MPa and a gaseous medium density of 25 kg/m^3. Nitrogen was used to fill the chamber to prevent self-ignition of the fuel, which distorts the pattern of atomization plume development. To analyze images, there are

For preliminary review

markers on the left side of each image, the distance between which is 50 mm. The structure and main elements of the atomization plume are shown in Fig. 5.176.

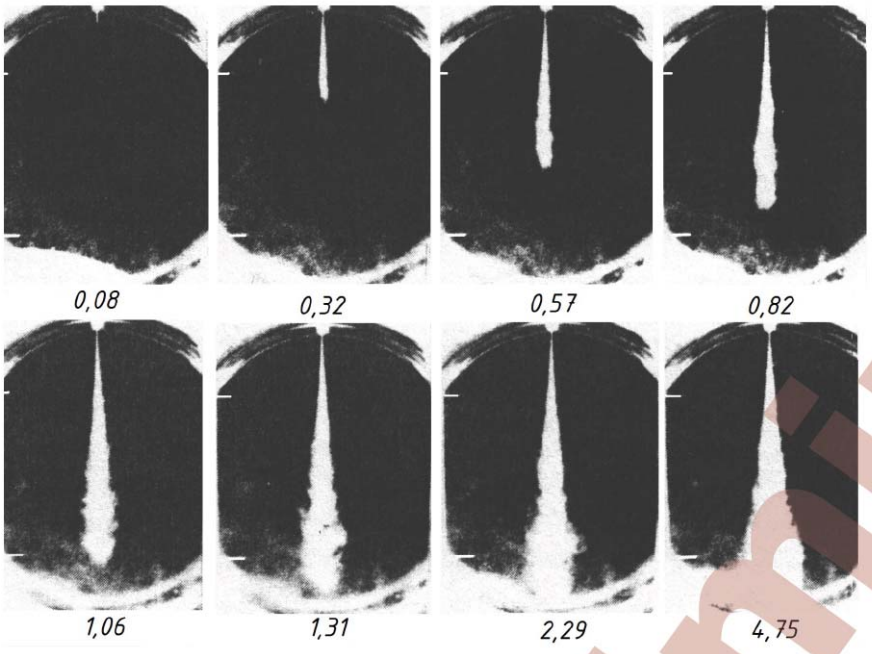


Figure 5.175 – Development of a liquid fuel atomization plume during one injection (the numbers indicate the time from the start of injection, ms) (adapted from [110])

Analysis of the images allows us to distinguish three main stages of the injection process:

- penetration of a homogeneous stream of fuel into the combustion chamber;
- fragmentation of the jet with the formation of an aerosol cloud;
- secondary disintegration of the atomization plume upon impact with the walls of the combustion chamber and the formation of a wall layer.

From the graph in Fig. 5.177, built from the results of processing the images presented in Fig. 5.175, it is clear that during the first 0.6 ms the front of the atomization plume propagates in a gaseous medium almost in the form of a homogeneous flow, called the *atomization plume core* (Fig. 5.176). The time before the start of its dispersed disintegration represents the period of penetration of the liquid core and determines the duration of the first stage (Fig. 5.176, Fig. 5.177).

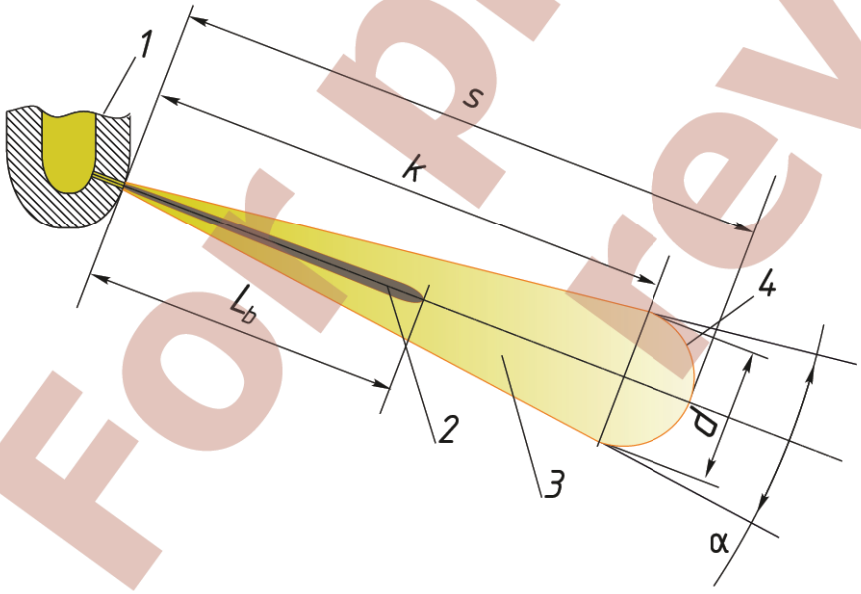


Figure 5.176 – Macrocharacteristics of a liquid fuel atomization plume: 1 – spray tip with fuel injector hole; 2 – liquid core of the atomization plume; 3 – aerosol shell; 4 – atomization plume front; L_b – depth of penetration of the atomization plume core before the start of its dispersed disintegration; k is the length of the atomization plume up to the largest cross section; s – atomization plume penetration depth; d – diameter of the atomization plume in the largest section; α is the cone angle of the atomization plume (adapted from [9])

When the fuel moves in the nozzle channel in the flow, a number of longitudinal and transverse disturbances arise, which are a consequence of turbulent pulsations and vortex formation in the flow due to the difference in velocities between its layers. The intensity of disturbances depends on the viscosity and compressibility of the fuel, the speed of the fluid in the fuel injector channel, the roughness of its walls and other design features of the hole itself.

For preliminary review

As the liquid core penetrates into the gaseous environment, disturbances in the flow intensify, and at some distance from the the nozzle, the core begins to actively disintegrate with small drops of fuel being separated from it. When the jet exits the fuel injector, it is affected by external factors – aerodynamic drag forces of the air charge, which deform and destroy the jet. These forces depend on the speed of fuel flow and the density of the air charge. They contribute to the separation of fuel particles from irregularities on the surface of the jet and their fragmentation into smaller droplets, as a result of which an aerosol shell begins to form around the core (Fig. 5.175). With increasing density of the working mixture in the cylinder, the influence of aerodynamic forces on fuel atomization increases.

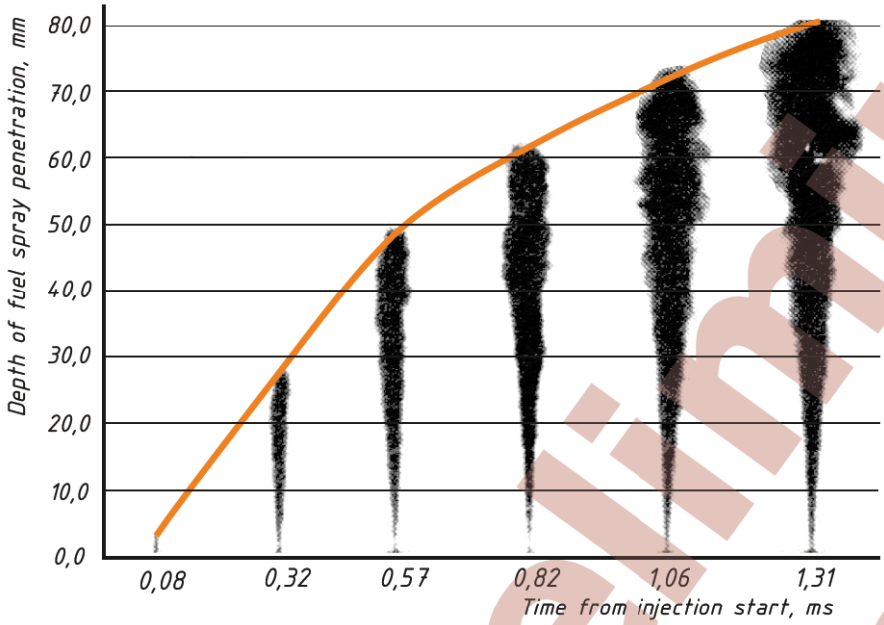


Figure 5.177 – Penetration depth of the atomization plume as a function of time from the start of injection (based on the results of processing photographs presented in Fig. 5.17) (adapted from [9])

The atomization of the jet is counteracted by the forces of surface tension and the forces of internal cohesion of fuel molecules, which tend to maintain its integrity. For this reason, the nature of the development of the atomization plume under the same injection conditions for different fuels will be different (Fig. 5.178). Reducing the viscosity of the fuel, for example by heating, weakens the effect of these forces and improves the quality of atomization.

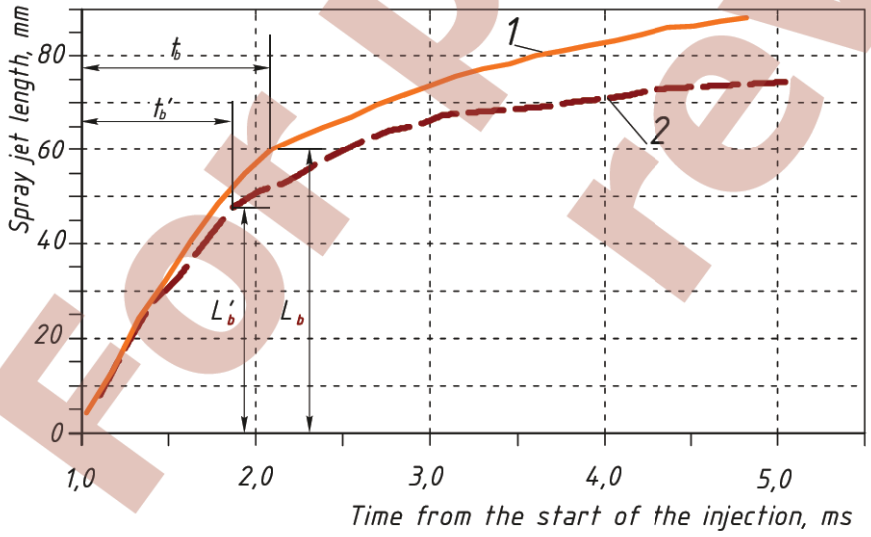


Figure 5.178 – Penetration depth of the atomization plume as a function of injection time for light (1) and heavy (2) motor fuel at atmospheric pressure (accumulator injection system with accumulator pressure of 100 MPa, fuel injector hole diameter 0.47 mm, control pulse duration 10 ms, fuel viscosity: light – 3 cSt, heavy – 12 cSt). (adapted from [111])

With the onset of dispersion decay of the core, the second stage of the fuel injection process begins, characterized by the formation of the main vortex, the front boundary of which is called the *atomization plume front* (Fig. 5.175).

For preliminary review

The figure (Fig. 5.177) shows that under given conditions, approximately 0.6 ms passes before the start of dispersive decay of the atomization plume core. Further, the size of the main vortex increases as the nozzle supplies new portions of fuel, and also as the main vortex penetrates into the gaseous environment. The last two frames show a fully formed atomization plume, the structure of which is shown in Fig. 5.175.

A real atomization plume does not have a clearly defined shape, therefore, to estimate its parameters, it is customary to consider the atomization plume as a cone with an apex at the fuel injector hole and a hemisphere at the base. The hemisphere limits the area corresponding to the atomization plume front. Based on this assumption, the main macroparameters of the atomization plume are (Fig. 5.176): the length of the liquid core, the penetration depth of the spray front, the cone angle and the volume of the space covered.

To find the depth of penetration of the atomization plume front and the period until the decay of its core, the dependencies proposed by can be used *Arai M.* and *M. Tabaia,* which make it possible to determine the current dislocation of the flame front based on the pressure values in front of the nozzle. In general, this dependence can be presented as follows:
for $0 < t < t_b$,

$$s = 0{,}39\left(\frac{2\Delta P}{\rho_f}\right)^{0{,}5} \times t, \tag{5.1}$$

for $t > t_b$,

$$s = 2{,}95\left(\frac{\Delta P}{\rho_a}\right)^{0{,}25} \times (d_n \times t)^{0{,}5}, \tag{5.2}$$

$$t_b = 28{,}65\frac{\rho_f \times d_n}{(\rho_a \Delta P)^{0{,}5}}.$$

To determine the distance from the tip of the nozzle to the localization of the atomization plume front at which the decay of the nucleus begins, the following relationship can be used:

$$L_b/d_n = 15.8(\rho_f/\rho_a)^{0.5},$$

where s is the localization of the atomization plume front; ΔP – the difference between the pressure in front of the nozzle and the back pressure in the combustion chamber; ρ_f – fuel density (for light fuels – 850 kg/m^3, for fuel oil – 920...1050 kg/m^3); t – time from the start of injection; ρ_a – charge density in the combustion chamber; d_n – diameter of the fuel injector hole; t_b – time before the start of decay.

Research has shown that, despite significant differences in physical properties, under the same conditions, various liquids are sprayed in a gaseous environment according to very similar patterns.

The depth of penetration of the atomization plume, together with the angle of its cone, determines the volume of capture of the combustion chamber space by the aerosol cloud (V_{cc}). Ultimately, macroparameters determine the value of the air-fuel ratio inside the aerosol cloud, on which the efficiency of the fuel combustion process largely depends.

As already noted, the aerosol plume does not have a clearly defined limiting surface, so there is no standard procedure for measuring its cone angle. One of the approaches to determining the angle of opening of the atomization plume is to find the arctangent of the ratio of the largest diameter of the atomization plume to the distance to the place of its dislocation (Fig. 5.180):

$$\alpha = \arctan(d/k),$$

where d is the largest diameter of the atomization plume; k is the distance from the tip of the fuel injector to the point where the largest diameter is measured.

The opening angle of the cone significantly affects the efficiency of air-fuel mixing: large angles indicate more intense air-fuel mixing, and small angles imply poorer mixing. Fuel-air mixing is proportional to the cross-sectional area of the atomization plume and approximately proportional to

For preliminary review

the cone opening angle.

Numerous studies have shown a strong dependence of the macroparameters of the atomization process on the injection pressure and the amount of back pressure in the combustion chamber. In Fig. 5.179 presents experimental data demonstrating the nature of changes in the macroparameters of the atomization plume for various injection pressures and back pressures in the combustion chamber when using light and heavy fuel.

Analysis of the dependencies shown in Fig. 5.179 shows that with increasing injection pressure, the range of the atomization plume increases in all cases. This is explained by an increase in the speed of liquid flow from the fuel injector hole, which largely depends on the pressure in front of the nozzle.

It should be noted that, all other things being equal, the macroparameters of fuel-air mixing of heavy fuel are worse than for light fuel.

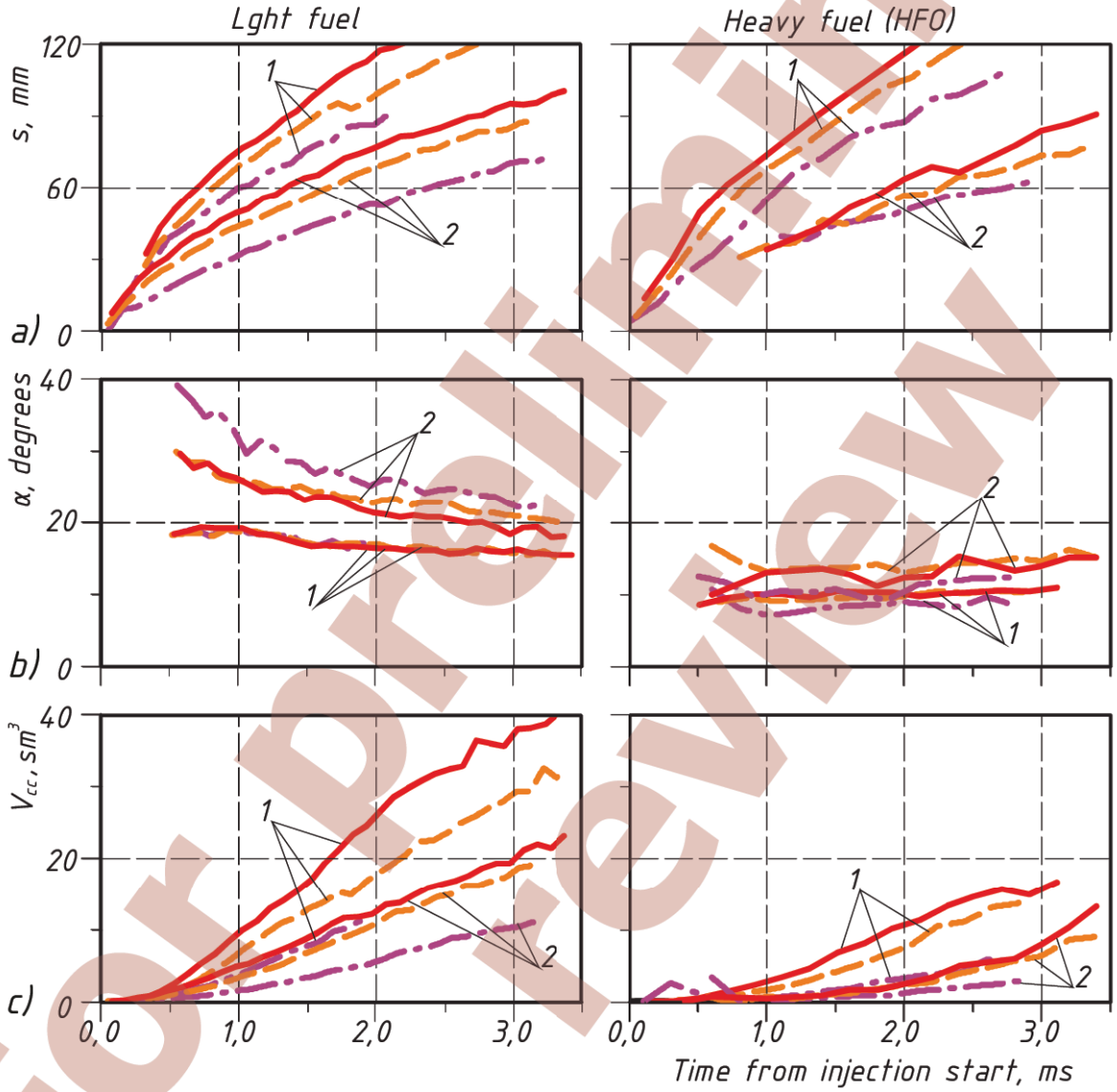


Figure 5.179 – Change in time of macroparameters of the atomization plume depending on the injection pressure for light and heavy fuel: 1 – back pressure 1.4 MPa; 2 – back pressure 5.0 MPa; *a* – depth of penetration of the atomization plume (*s*); *b* – taper of the atomization plume (α); *c* – volume of space coverage by the atomization plume (V_{cc}): ——— – injection pressure 140 MPa; – – – – injection pressure 100 MPa; – · – · – – injection pressure 60 MPa (adapted from [111])

It is clearly seen that the magnitude of the backpressure has a significant influence on the macroparameters of the injection process. If for light fuel at low back pressures the taper value is practically independent of the injection pressure, then with an increase in back pressure there is a decrease in the taper angle with increasing injection pressure. For heavy fuels, the taper angle is

For preliminary review

smaller and varies within very small limits, remaining almost constant throughout the entire injection. However, the value of the atomization plume opening angle in all cases increases with increasing backpressure, which is explained by a decrease in the penetration depth of the liquid core of the atomization plume before its dispersed disintegration begins.

A smaller penetration depth and plume opening angle lead to the fact that the volume coverage of the combustion chamber for heavy fuels will be lower than in the case of light fuels.

The difference in atomization macroparameters for different fuels is explained mainly by differences in their viscosity (for the case under consideration, 3 cSt for light fuel versus 12 cSt for HFO).

In an engine, the process of fuel atomization in the diesel combustion chamber is accompanied by its evaporation, and at a certain stage, combustion. In Fig. 5.180 shows the dependences of the penetration of the atomization plume front into the gaseous medium, taking into account the evaporation of fuel for three pressure values in front of the nozzle.

Injection into the experimental chamber was carried out using an accumulator system at constant pressure. The back pressure in the chamber was maintained at 2.6 MPa, the temperature was 650 K, and the duration of the control signal supplied to the fuel injector was 10 ms.

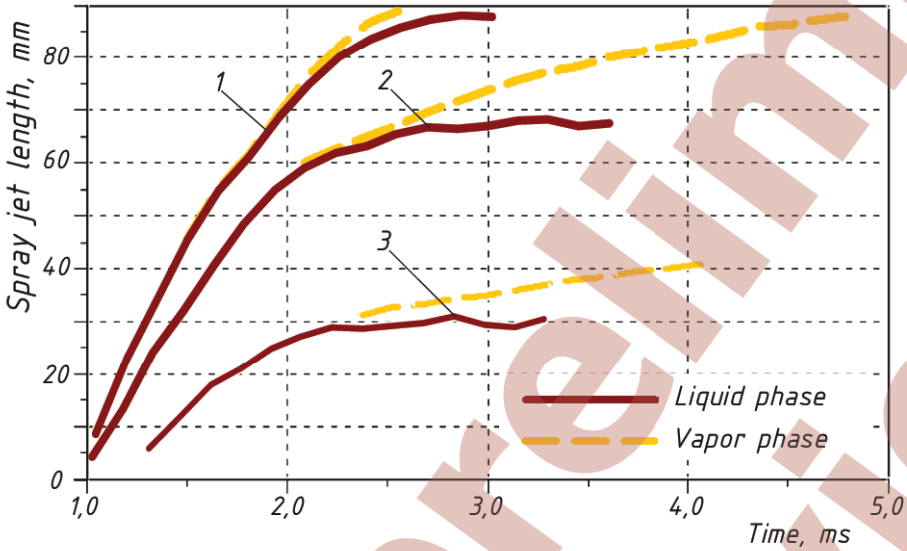


Figure 5.180 – Depth of penetration of the liquid and vapor phase of fuel into the combustion chamber for a pressure in front of the nozzle of 140 MPa (1), 100 MPa (2), 60 MPa (3) (adapted from [111])

From the presented dependences it is clear that at some distance from the tip nozzle the entire liquid phase completely evaporates. If this occurs before the atomization plume reaches the walls of the combustion chamber, the injection process is limited to two stages.

If, by the time it reaches the walls of the combustion chamber, the speed of the atomization plume remains sufficiently high, and the evaporation process has not yet been completed, the aerosol cloud flows onto the walls of the combustion chamber. In this case, the leading front of the atomization plume is deformed, part of the fuel bounces off the walls, and part settles on the walls in the form of a thin film (Fig. 5.185). The third stage begins, the stage of secondary decomposition and formation of a near-wall layer of the fuel film.

The boundaries of the atomization plume are shown in Fig. 5.181 lines. Each subsequent line characterizes the position of the atomization plume front with a time interval of approximately 0.01 ms. The figure shows that for the case under consideration, the atomization plume will reach the piston bottom approximately 1.5 ms after the start of injection. At the same time, studies on the volatility and flammability of liquid fuels indicate that the liquid phase concentrated in the atomization plume will reach the piston walls even before the fuel has time to completely evaporate. In this regard, the secondary disintegration of the atomization plume from impact on the surface of the combustion chamber has a significant impact on the efficiency of fuel combustion. As a result of the impact, the front of the atomization plume is deformed, some of the drops change the direction of their movement, and some settle on the surface of the combustion chamber in the form of a thin film.

For preliminary review

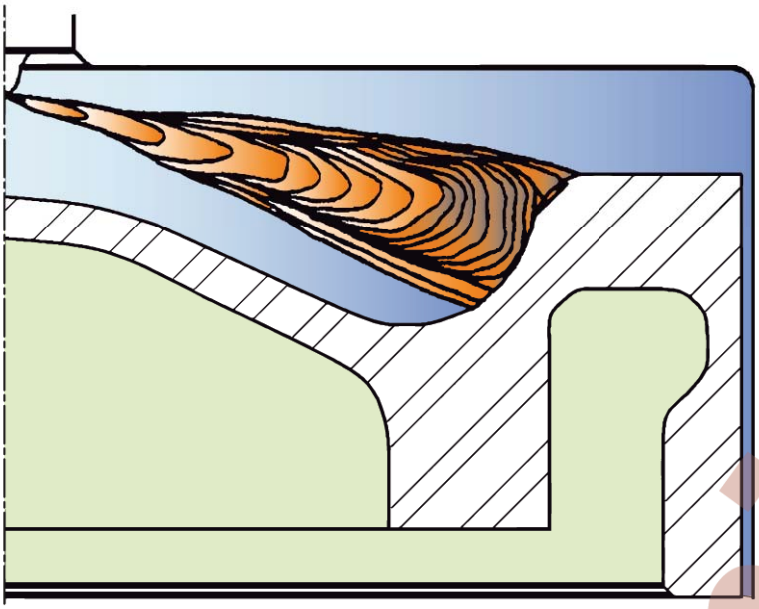


Figure 5.181 – The third stage of the injection process – the inflow of the atomization plume onto the walls of the combustion chamber, accompanied by secondary disintegration of the atomization plume from hitting the wall, and the formation of a wall layer (adapted from [110])

5.9.3 Methods of mixture formation in marine diesel engines

The nature of the engine operating process significantly depends on the selected type of mixture formation. Marine diesel engines are characterized by direct (unseparated) mixture formation, in which direct fuel injection is carried out into the combustion chamber limited by the fire bottom of the cylinder cover, the cylinder walls and the fire bottom of the piston. Depending on the design features of the engine and approaches to organizing its operating process, unseparated mixture formation can be implemented as: volumetric, volumetric-film and film (Fig. 5.182).

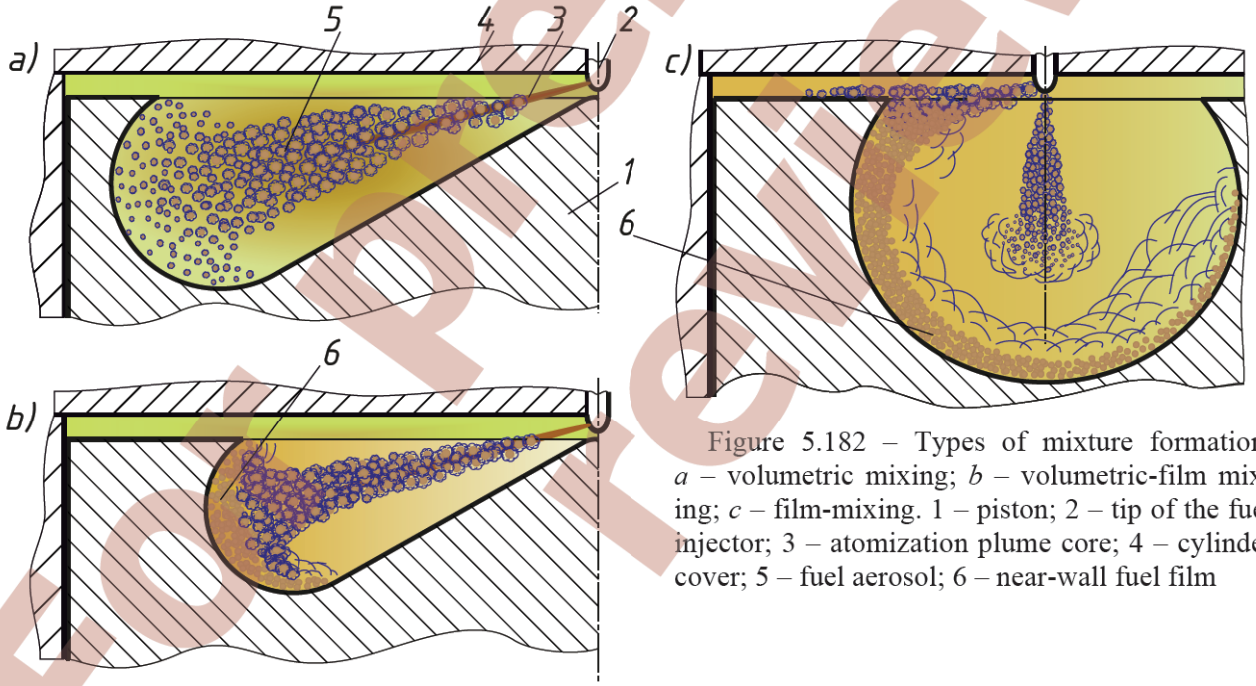


Figure 5.182 – Types of mixture formation: *a* – volumetric mixing; *b* – volumetric-film mixing; *c* – film-mixing. 1 – piston; 2 – tip of the fuel injector; 3 – atomization plume core; 4 – cylinder cover; 5 – fuel aerosol; 6 – near-wall fuel film

Volumetric mixture formation is characterized by the fact that 90...95% of the fuel that is injected into the volume of the combustion chamber turns into an aerosol and evaporates during the movement of droplets from the tip of the fuel injector to the walls of the combustion chamber (Fig. 5.182 *a*). This is ensured by fairly high fuel injection pressures, which for LSE are in the range of 60...95 MPa, and for MSE and HSE 100...220 MPa and more. Such pressures make it possible to obtain sufficiently small aerosol droplets that can warm up and evaporate before the front of the atomization plume reaches the walls of the combustion chamber. To maximize the full coverage of

For preliminary review

the combustion chamber volume with fuel aerosol, on engines with volumetric mixture formation, multi-hole injectors with the number of fuel injector holes from 5 to 18 are used. Obviously, the completeness of fuel evaporation significantly depends on the distance traversed by the front of the atomization plume to the walls of the combustion chamber. For this reason, volumetric mixture formation is realized in combustion chambers that have a shallow depth and large diameter (the ratio of the combustion chamber diameter to the cylinder diameter: d_{cc}/D = 0.75...0.85 or even 1.0 for two-stroke LSEs), which makes it difficult to use of this type of mixture formation in small engines. In addition to the quality of atomization and the size of the combustion chamber, the efficiency of mixture formation is significantly influenced by the presence of organized vortex formation of charge flows in the combustion chamber. Therefore, when designing an engine, additional conditions are created to impart mobility to the charge. A significant role in this process is played by fuel atomization, in which flows of fuel droplets transfer their kinetic energy to the charge, drawing it into motion.

Despite the listed factors, with this type of mixture formation there is a large uneven distribution of fuel throughout the volume of the combustion chamber. Therefore, high combustion completeness during volumetric mixture formation can be achieved only with large values of the excess air coefficient α. At nominal mode for MSE and HSE, its value should be within the range α = 1.6…2.2, for two-stroke low-speed diesel engines α = 2.5…3.5.

The volumetric method of mixture formation provides high efficiency at a high rate of pressure increase ($dp/d\varphi$ = 0.6...1.0 MPa/deg), and the operating process is characterized by high maximum combustion pressures p_z, which leads to increased loads on the crankshaft and accompanied by quite noisy engine operation. Despite this, in most marine diesel engines, main and auxiliary, it is volumetric mixture formation that is implemented.

The design of the combustion chamber for the implementation of volumetric mixture formation significantly depends on the type of engine and its layout. It should provide:

- high degrees of purification from exhaust gases and filling with fresh charge;
- reduction of combustion chamber surface ratio F_{cc} to its volume V_{cc}, which reduces heat losses into the cooling system;
- optimal degree of turbulence at the inlet and compression;
- reduction of combustion duration.

In case of volumetric mixture formation, the atomization plume must ensure uniform distribution of aerosol droplets throughout the entire volume of the combustion chamber. To do this, the shape of the combustion chamber, the location of the fuel injector(s), as well as the shape and direction of the atomization plumes are coordinated with each other by selecting the geometry of the nozzle holes.

Modern low-speed marine engines are characterized by the use of combustion chambers, most of which are located on the cylinder cover, and the smaller part in the piston (Fig. 5.183 *a*). This design makes it possible to increase the area of the fire surface of the cover, from which it is relatively easy to remove heat, and to reduce the area of the fire surface of the piston, thereby reducing the amount of heat removed and the thermal stress of its bottom. For two-stroke engines, in which heat is generated at every revolution of the crankshaft, the problem of cooling the pistons is especially pressing.

The part of the combustion chamber located in the cylinder cover has a vaulted shape, tapering towards the exhaust valve seat.

For engines with cylinder diameters up to 50 cm, in which two fuel injectors are usually installed in the cylinder cover, the piston bottom is made in the form of a spherical recess (Fig. 5.176 *a*, left). This shape allows the atomization plume to be dispersed in space to avoid their overlap (Fig. 5.183 *b*), to more evenly distribute the load on the piston bottom, with a minimal increase in the fire bottom area.

For engines with cylinder diameters larger than 50 cm, the combustion chamber is usually equipped with three fuel injectors; therefore, to prevent overlap of plumes from different fuel injectors, they are directed peripherally (Fig. 5.183 *c*). As a result, a zone not covered by plumes is

For preliminary review

formed in the central part of the combustion chamber, in which the air is not used efficiently. To reduce the volume of the uncovered zone, a displacer is placed in the central part (Fig. 5.183 *a*, right), which increases the concentration of air in the penetration zone of the atomization plumes and reduces direct contact of fuel with the piston. This design was called Oros, which translated from Greek means hill, small slide.

On medium-speed and some high-speed marine engines, of all the variety of forms, the most widely used is the «Hesselmann» type chamber, which maximally follows the contours of the atomization plume with a central location of the multi-hole fuel injector (Fig. 5.184 *b*). Such combustion chambers are located primarily in the piston, but a small part of the volume can be placed in the cylinder cover (Fig. 5.184 *c*). The central displacer on the bottom of the piston helps push air from areas not covered by plumes into the area of their formation and penetration, thereby improving the use of oxygen contained in the air.

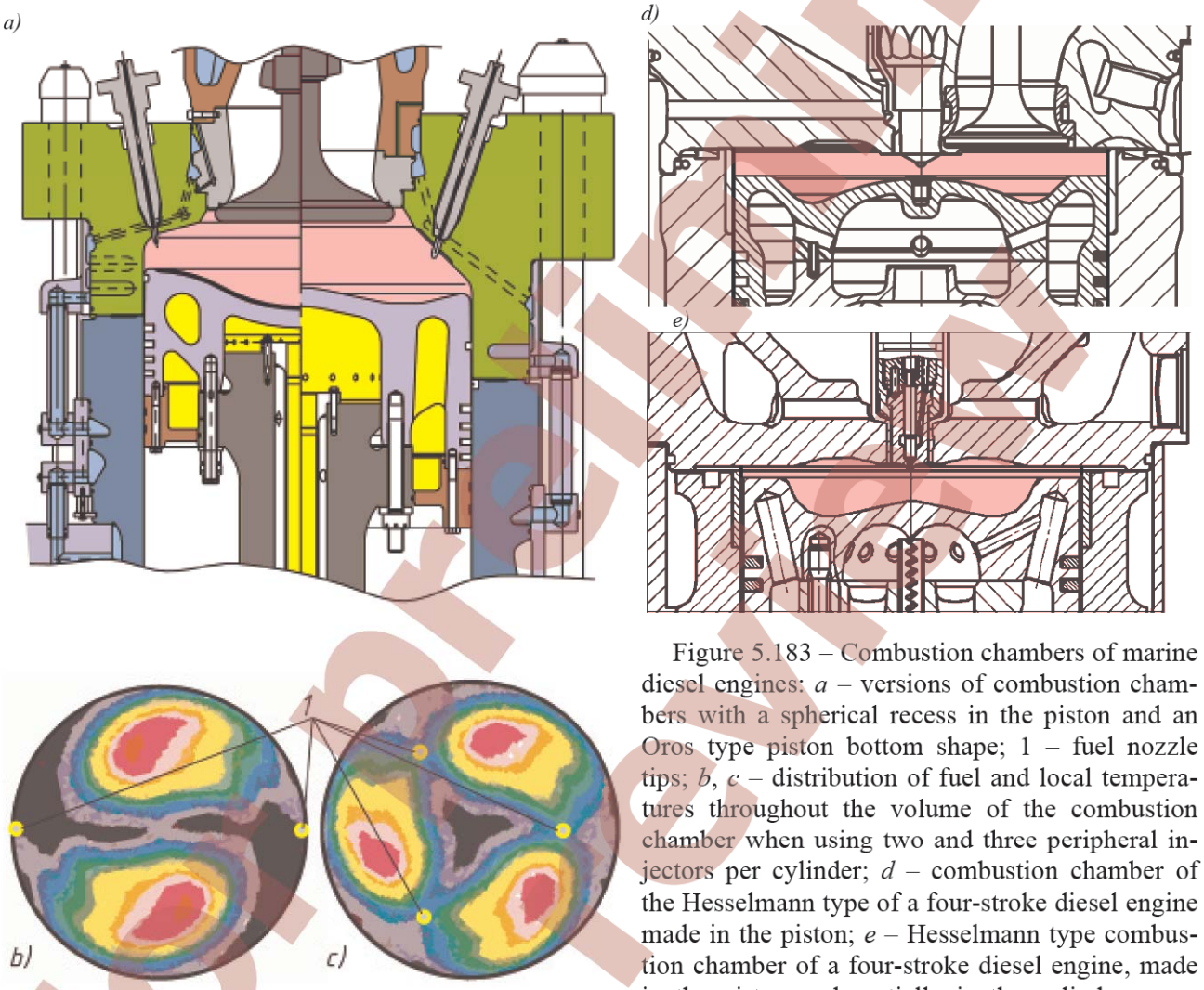


Figure 5.183 – Combustion chambers of marine diesel engines: *a* – versions of combustion chambers with a spherical recess in the piston and an Oros type piston bottom shape; 1 – fuel nozzle tips; *b*, *c* – distribution of fuel and local temperatures throughout the volume of the combustion chamber when using two and three peripheral injectors per cylinder; *d* – combustion chamber of the Hesselmann type of a four-stroke diesel engine made in the piston; *e* – Hesselmann type combustion chamber of a four-stroke diesel engine, made in the piston and partially in the cylinder cover (adapted from [22, 112])

The quality of volumetric mixture formation significantly depends on the presence and degree of air charge turbulization. In two-stroke diesel engines, the vortex motion of the charge is additionally created at the purge stage due to the inclined arrangement of the purge ports in the cylinder liner. For each specific case, there are optimal speeds and directions of charge movement, when deviating from which aerosol droplets and fuel vapors from the volume of one atomization plume can be transferred by the movement of the charge into the volume of another atomization plume, leading to the formation of local zones with a low fuel-air ratio and, as a result, deterioration mixture formation.

For preliminary review

A type of volumetric mixture formation can be considered a layer-by-layer method of forming a fuel-air mixture, in which the cyclic supply is divided into several portions entering the working cylinder sequentially. To form a layer-by-layer charge, reusable injection is used within one cycle. Usually two or more sequential injections are used. Each portion serves to have a specific impact on the nature of the operating process. The first portion supplied by the fuel injector is 5...15% of the total cyclic supply and is called pilot or ignition. It serves to heat the charge and form open flame, which accelerates the process of pre-flame processes of the main portion of fuel, which is supplied to the combustion chamber with some delay. Due to the low supply, the mixture formation of the pilot portion is mainly volumetric in nature, so all pre-flame processes occur quite quickly. The advance of the supply of the pilot portion in relation to the main one is selected in such a way that by the time the main portion is injected, a source of open flame is already present in the cylinder. Ignition and combustion of pilot fuel, due to its small amount, does not lead to a significant increase in pressure on the compression line, but it significantly increases the temperature and creates a source of open flame. The main portion of fuel injected into the environment prepared in this way burns as it enters through the nozzle, without a sharp increase in pressure. As a result, the rigidity of the operating process is significantly reduced.

The method of layer-by-layer mixture formation is widespread in modern engines, especially with electronic control of the fuel supply process.

We will consider in more detail the methods of organizing layer-by-layer mixture formation in the following sections.

Volumetric-film mixture formation. This mixture formation is characterized by the fact that approximately 40...55% of the droplets of atomized fuel reach the walls of the combustion chamber, and the remainder is distributed throughout its volume (Fig. 5.182 *b*). This type of mixture formation is typical for engines with direct fuel injection and having small cylinder diameters with a ratio of d_{cc}/D = 0.50...0.60. It is implemented in high-speed marine diesel engines, most often used as emergency ones.

To implement this method of mixture formation, injectors are usually used, having from 3 to 8 fuel injector openings supplying fuel under a pressure of 120...180 MPa. The fuel, distributed throughout the volume of the combustion chamber, interacting with the hot charge is the first to undergo pre-flame stages. The fuel that gets on the walls spreads over the surface of the chamber in the form of a film 0.01...0.025 mm thick, warms up, evaporates and mixes with the vapors produced in the volume of the combustion chamber.

The presence of a near-wall layer significantly slows down the rate of mixture formation and combustion due to lower film temperatures, which reduces the rate of fuel evaporation and mixing of its vapors with air. As a result, the rate of heat release is reduced, especially during the initial period of combustion. Once combustion begins, an increase in temperature in the combustion chamber leads to an increase in the rate of evaporation and mixing, which leads to a significant increase in the rate of heat release.

For effective evaporation of the near-wall layer of fuel, the temperature of the walls of the combustion chamber must be sufficiently high (330...350°C), but not sufficient for thermal decomposition of the fuel with the formation of solid deposits. It is for this reason that the combustion chambers with this method of mixture formation are located exclusively in the piston (Fig. 5.184), the fire bottom of which is the hottest surface.

The quality of volumetric-film mixture formation significantly depends on the presence of charge turbulization, which is achieved by displacing air from the narrowest places in the above-piston space during the compression process (Fig. 5.185). The improvement of mixture formation is also facilitated by the vortex movement of the charge at a speed of 25...30 m/s along the cylinder axis, which is formed during the intake stroke due to the tangential arrangement of the intake valve channels.

This type of mixture formation ensures a fairly efficient flow of the operating process, with high levels of power and environmental friendliness with a moderate rate of pressure increase ($dp/d\varphi$ = 0.25...0.4 MPa/deg).

For preliminary review

The shape of combustion chambers during volumetric-film mixture formation is very diverse (Fig. 5.184), however, on marine engines, ω- shaped combustion chambers (Fig. 5.184 *a*, *b*), truncated conical (Fig. 5.184 *c*) and toroidal (Fig. 5.184 *d*, *e*).

Often, to facilitate maintenance, in small engines the fuel injector is positioned obliquely and removed from under the valve box of the cylinder cover (Fig. 5.184 *b*, *c*, *d*, *e*).

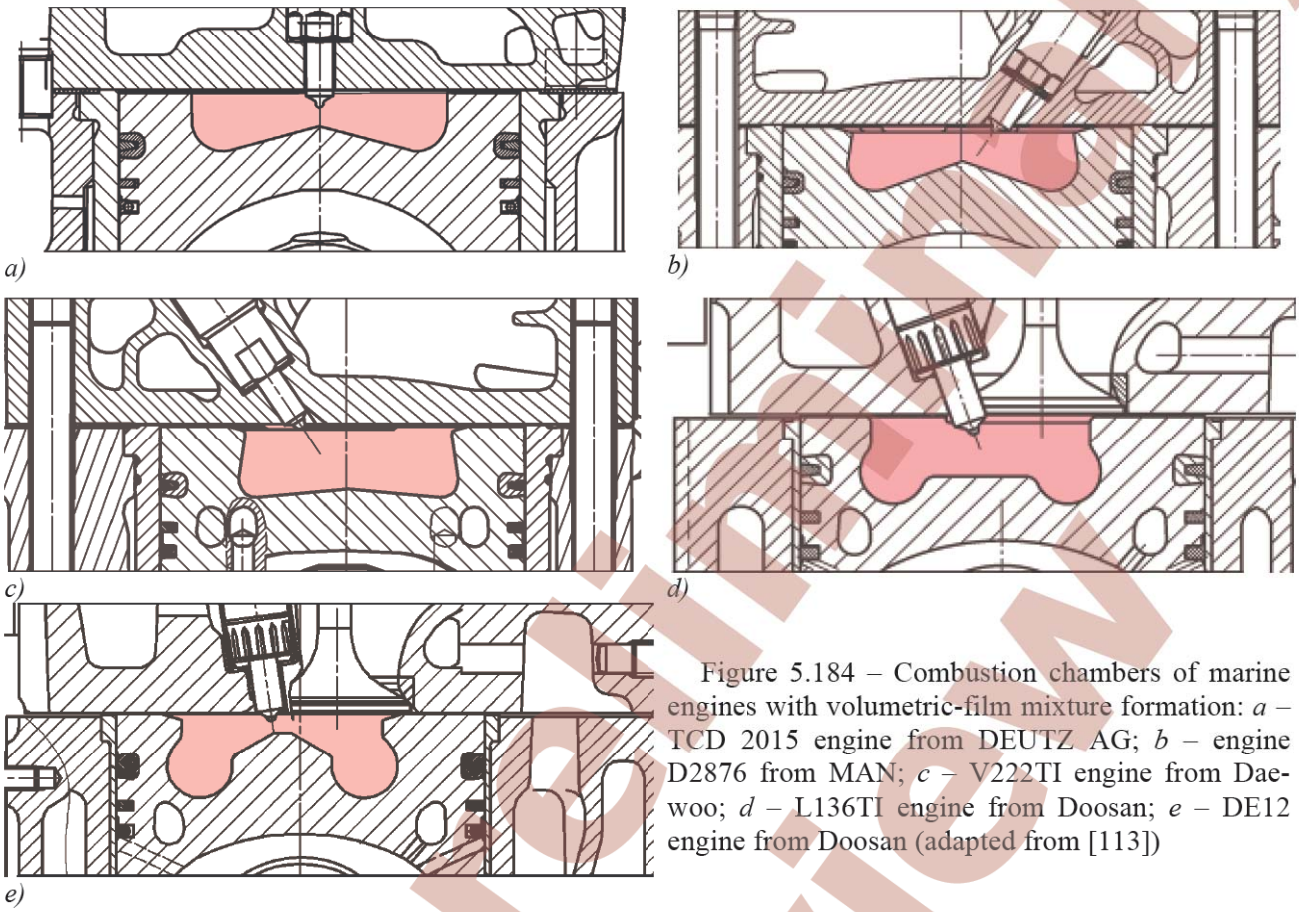


Figure 5.184 – Combustion chambers of marine engines with volumetric-film mixture formation: *a* – TCD 2015 engine from DEUTZ AG; *b* – engine D2876 from MAN; *c* – V222TI engine from Daewoo; *d* – L136TI engine from Doosan; *e* – DE12 engine from Doosan (adapted from [113])

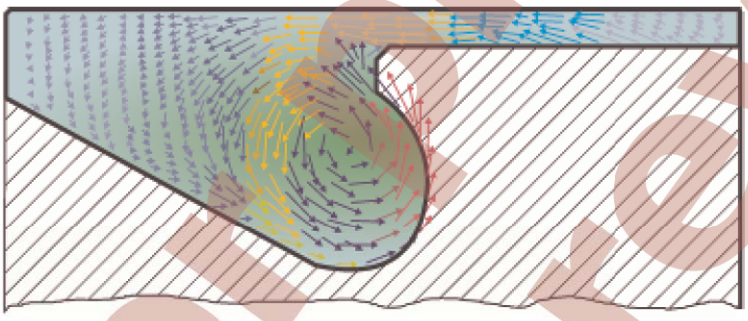


Figure 5.185 – Charge turbulization during the compression process due to the displacement of air from the narrowest places in the above-piston space

Film mixture formation is characterized by the fact that the main part of the fuel (90...95%) is supplied by a fuel injector to the hot walls of a spherical combustion chamber located in the piston, where a thin fuel film is formed, which quickly warms up and evaporates, taking away some of the heat from the walls. The remaining part of the fuel is sprayed by a fuel injector in the volume of the combustion chamber (Fig. 5.182 *c*). This part of the fuel quickly passes through the pre-flame stages and ignites, forming open flame sources that serve to ignite the combustible mixture that forms above the film.

This method provides a relatively low rate of pressure increase ($dp/d\varphi$ = 0.2...0.3 MPa/deg) and good efficiency, but requires maintaining the piston temperature within specified limits, ensuring intensive evaporation of the fuel film. Film mixture formation has practically not been used on marine engines, since for such diesel engines it is difficult to boost their power, in particular by using supercharging.

For preliminary review

5.9.4 Microscopic parameters of the injection process, their influence on mixture formation and combustion in diesel engines

The processes of mixture formation and combustion of fuel in diesel engines have a relatively short period of time, which ranges from 0.005 seconds for HSE to 0.05 seconds for LSE. During this period, the fuel introduced by the fuel injector must be evenly distributed throughout the volume of the combustion chamber, warm up, evaporate, and only then ignite and burn. In order for all of the above processes to be completed in such a short period of time, the fuel must be sprayed into tiny droplets. This increases the contact surface of the droplets with hot gases and reduces the duration of the evaporation process.

According to experimental data, a jet of atomized fuel consists of droplets, the average diameter of which for low-speed diesel engines is 15...25, for high-speed diesel engines 5...10 μm. With such a fine spray, the number of droplets can reach a huge number. So, for example, with a cyclic supply to the LSE equal to 85 cm^3 (3000 kW/cyl. at 120 min^{-1}), fuel atomization to 20 μm leads to the formation of more than 2.5 billion droplets with a total surface area of more than 12.6 m^2.

The main microparameters characterizing the quality of fuel atomization are the fineness or dispersion of the atomization, assessed by the average diameter of the fuel droplets, and uniformity, assessed by the degree of deviation of the droplet sizes from the average value.

With a decrease in cylinder diameter and an increase in diesel engine speed, the requirements for fineness and uniformity of atomization increase.

To assess the quality of atomization, the arithmetic average d_{10}, surface d_{20} and volumetric d_{30} droplet diameters are most often used, but the most common criterion for assessing the average diameter of fuel droplets is the technique proposed by Josef Sauter in 1926. The average diameter according to Sauter (d_{32}) is taken to be the size at which the ratio of volume to surface for the totality of all measured drops is equal to the same ratio of drops of average size.

To estimate d_{32} from measurement results, the following relationship is used :

$$d_{32} = \left(\sum_i N_i D_i^3 \right) / \left(\sum_i N_i D_i^2 \right).$$

The size of the droplets and their uniformity have the most significant impact on the flammability and completeness of fuel burnout.

Currently, two methods are used to determine the fineness of fuel atomization. The first, classic, using a smoked record. The glass plate is covered with soot over a kerosene torch to a thickness of 50...100 μm, then covered with a thin layer of magnesium combustion products. The plate sweeps once past the measurement zone near the operating fuel injector. The plates are photographed through an optical microscope in reflected light, and then the resulting negatives are processed.

The sooty plate method has remained the standard for decades in terms of reliability, versatility, relative accuracy, simplicity and low cost. It can be used for any fuels and backpressures, but is characterized by significant complexity in processing experimental data.

The availability of modern scanning and programming tools eliminates the most significant drawback of the method – the labor intensity of processing.

As an alternative to the sooty plate method, the laser diffraction method (small-angle scattering), based on the difference in light diffraction when passing by droplets of different sizes, is increasingly being used. The method becomes one of the main ones when studying the dispersion and concentration of sprayed media. The measurements are not sensitive to the speed of movement and distribution of droplets in the flow, physicochemical or other properties of the sprayed liquid.

Based on the results of measuring the size of aerosol droplets, to assess the quality of atomization in terms of fineness and uniformity, characteristics are built that show the nature of the particle size distribution.

In Fig. 5.186 *a* shows the dependence of the atomization quality for three fuel injectors with different numbers of nozzle holes with the same total flow area. Figure 5.186 *b* shows the atomization characteristics for light and heavy fuels under the same initial conditions.

The closer the vertex of the atomization characteristic is located to the ordinate axis and the nar-

For preliminary review

rower the range of droplet diameters in which the characteristic is located, the finer and more uniform the atomization it displays.

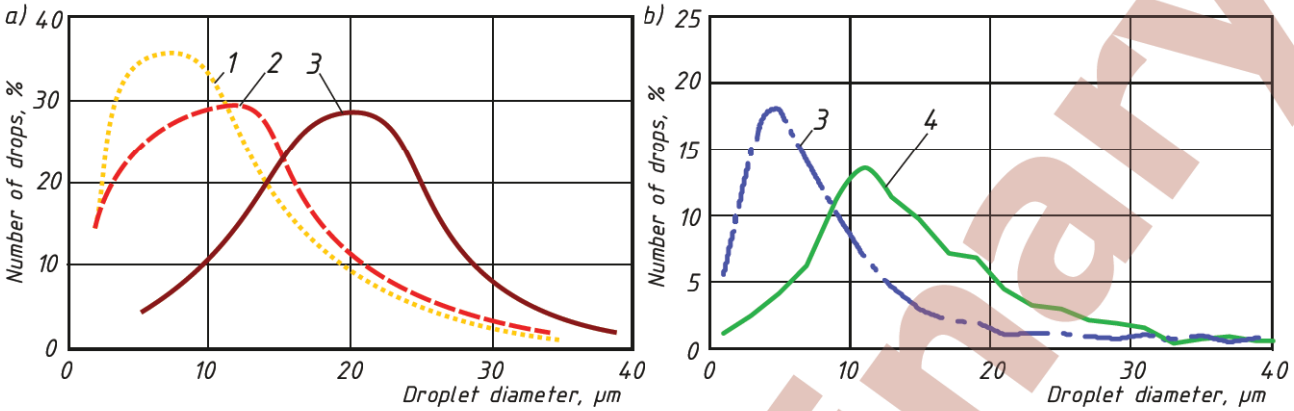


Figure 5.186 – Characteristics of fineness and uniformity of fuel atomization: *a* – for one total flow area with different diameters and numbers of nozzle holes; 1 – four nozzle holes with diameter 0.4 mm; 2 – two nozzle holes with diameter 0.57 mm; 3 – one nozzle hole with diameter 0.8 mm; *b* – under the same initial conditions for heavy and light motor fuel (nozzle hole 0.27 mm, pressure in front of the fuel injector 100 MPa, back pressure 1.4 MPa). 4 – heavy fuel (12 cSt); 5 – light fuel (3 cSt) (adapted from [9])

It should be understood that the average diameter of aerosol droplets is a very conditional parameter, since under the influence of a number of factors it can change both in time and depending on the location of its measurement along the length of the plume.

In table 5.9 presents the results of measurements d_{32} under various injection conditions, obtained for a accumulator-powered fuel supply system with a accumulator located in the injector housing.

The injection was controlled using an electronic system. The injection duration in all cases was 5 ms.

Table 5.9 – Results of measurements of the average droplet diameter (d_{32}) of light motor fuel under various injection conditions

Diameter holes, mm	Pressure in front of the nozzle, MPa	Density gas environment, kg/m^3	Localization measurements, mm	Average diameter according to Sauter, μm
0.4	83	25	120	25.40
0.2	83	25	120	23.28
0.4	28	25	120	39.19
0.4	55	25	120	27.10
0.2	83	1.2	120	12.67
0.2	83	25	60	12.39
0.2	83	25	80	16.33
0.2	83	25	100	20.96

The patterns of particle formation and distribution are the result of the combined action of several factors, which together determine the characteristics of an aerosol. The main factors include: injection pressure, distance from the tip of the nozzle to the measurement localization, density of the gas medium, diameter and length of the nozzle hole of the nozzle, needle lifting speed, etc.

Changes in the average size and velocities of aerosol droplets over time. In Fig. 5.187 shows the time dependence of the change in the average diameter and the velocity distribution of aerosol droplets in the centre of the atomization plume during the injection of light fuel by the accumulator fuel supply system of a high-speed diesel engine. The dependences were obtained for dislocation

For preliminary review

measurements 20 mm from the fuel injector hole by processing 20 thousand measurements obtained over 200 consecutive injections.

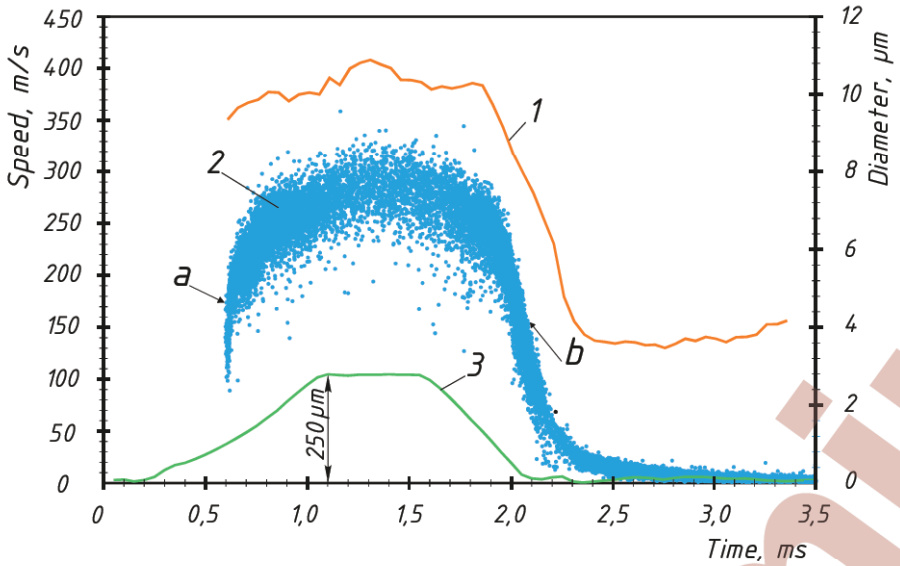


Figure 5.187 – Change in the average diameter and speed of aerosol droplets as a function of time from the start of injection: 1 – average droplet diameter; 2 – drop speed; 3 – lifting the needle valve of the nozzle. *a* – formation of the atomization plume front; *b* – tail of the atomization plume (adapted from [114])

From the presented dependencies it is clear that at the beginning of injection, the velocities of aerosol droplets vary widely from 80 to more than 300 m/s. The highest droplet velocities are observed after the needle valve is fully opened and remain virtually unchanged throughout the entire injection period. At the end of the injection, the droplet velocity decreases rapidly.

The distribution of the average diameter of droplets and their average speed in a plane perpendicular to the axis of the high-speed diesel atomization plume is shown in Fig. 5.188. Average values are determined for the time interval corresponding to full needle elevation. The largest droplets with the highest speed are located along the axis of the atomization plume. As the droplets move away from the axis, the velocity of the droplets and their average diameter decrease significantly. If in the centre of the atomization plume the speed reaches 260 m/s, then already at a distance 1.5 mm from the axis the speed drops to 20 m/s. Accordingly, the average diameter decreases from 10 μm in the centre to 4 μm at the atomization plume generatrix.

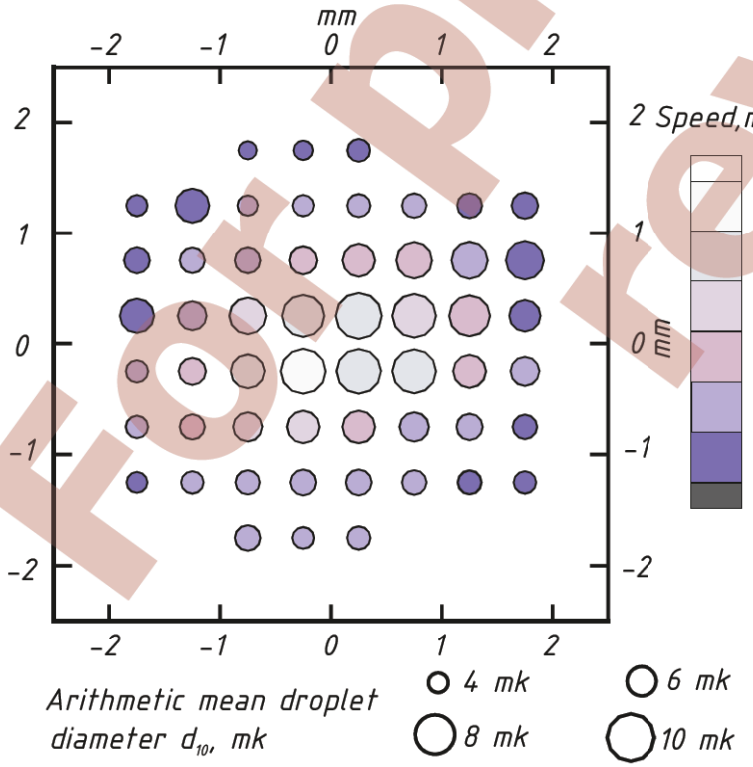


Figure 5.188 – Distribution of the average diameter of droplets and their average speed in a plane perpendicular to the axis of the high-speed diesel atomization plume (adapted from [114])

For preliminary review

Some asymmetry of the atomization plume in a plane perpendicular to its axis is explained by the throttling effect of the shut-off cone of the needle valve, which creates unequal conditions for the entry of the liquid flow into the fuel injector hole channel.

The influence of air charge density on the quality of atomization. The reason for the separation of droplets by speed is that the fuel jet enters a low-moving gaseous environment. As a result of the exchange of momentum, the front of the atomization plume slows down, and the surrounding air, penetrating between the aerosol particles inside the atomization plume, accelerates. As a result, drops located in the front of the plume are decelerated more strongly, and those located behind the front of the plume are slower. This leads to an increase in the concentration of aerosol droplets in areas adjacent to the atomization plume front, the greater the concentration of aerosol droplets, the deeper the plume penetrates.

At low back pressures, the change in the average droplet diameter over time practically repeats the dependence of the change in the average droplet velocity. The smallest droplets have the slowest speed, and vice versa.

However, as the back pressure increases, there is a significant effect of air density on the speed and size of the droplets (Fig. 5.189).

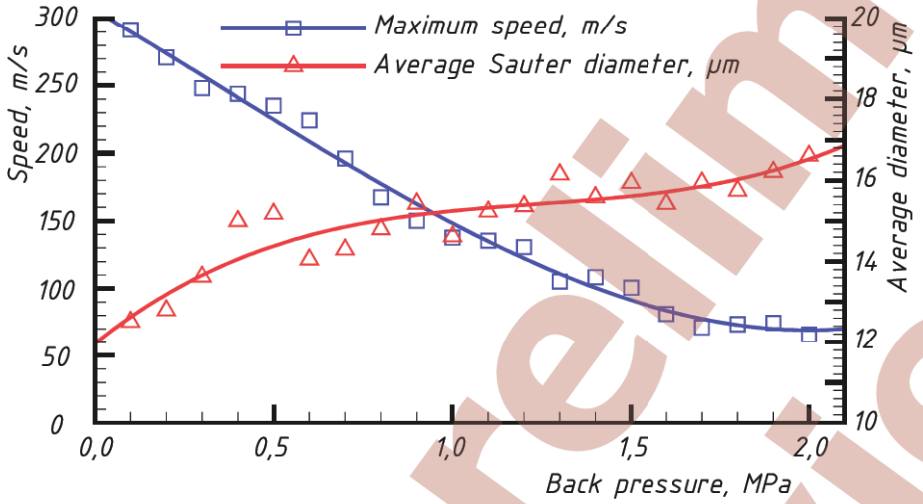


Figure 5.189 – Effect of back pressure on the maximum speed of propagation of the atomization plume front and the average diameter of aerosol droplets (adapted from [114])

With increasing density, the mechanism of interaction of the atomization plume with the gaseous medium changes, as a result of which the effect of back pressure on the atomization fineness is ambiguous. On the one hand, the higher the density, the stronger the friction between the gas and the liquid flow. Therefore, the velocity difference between the layers of liquid increases faster, which leads to an earlier disintegration of the atomization plume into individual droplets. As a consequence of this, the length of the atomization plume decreases, and the fineness of the spray increases. On the other hand, a denser medium promotes the stratification of aerosol droplets by speed and an increase in the frequency of collisions.

A gas environment with a higher density slows down the penetration of the aerosol and significantly reduces the distance before the liquid core begins to disintegrate. As a result, droplets form earlier and travel a greater distance. These two factors – lower speed and longer penetration distance – cause small droplets to join together to form larger ones. Thus, with increasing back pressure, the average droplet diameter increases.

Change in the average particle size along the length of the atomization plume. In Fig. 5.190 shows the change in d_{32} of drops of light fuel at the boundary of the atomization plume front for axial localizations of measurement from 40 to 140 mm from the tip of the nozzle. The dependences were obtained in an experimental chamber for an injection pressure of 83 MPa, a nitrogen density of 25 kg/m^3, and a fuel injector hole diameter 0.2 mm (Table 5.9). From Figure 5.190 it is clear that d_{32} drops increases as the atomization plume front penetrates into the gaseous medium.

For light diesel fuel, the relationship between measurement location and average diameter is almost linear.

For preliminary review

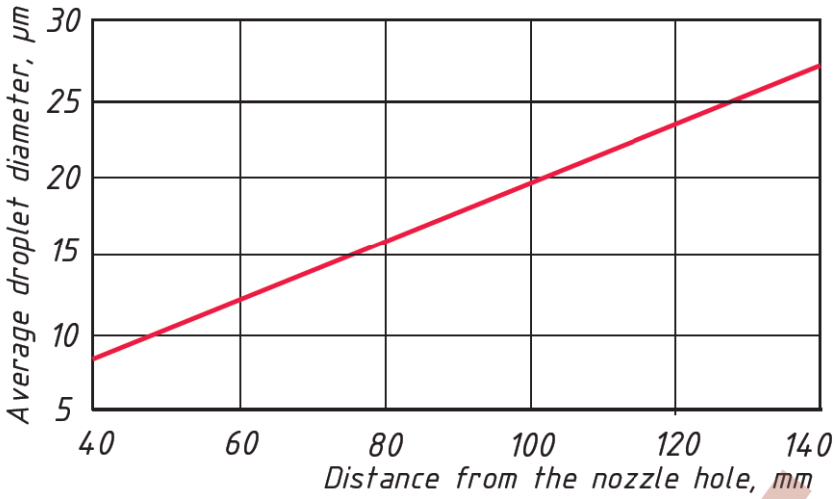


Figure 5.190 – Effect of measurement localization on the average droplet diameter (d_{32}) of diesel fuel aerosol (diameter of the fuel injector opening 0.2 mm; gas density 25 kg/m^3; pressure in front of the nozzle 83 MPa) (adapted from [9])

The increase in droplet sizes along the length of the atomization plume is explained by a number of reasons:

- large drops carry a higher impulse and, accordingly, have greater inertia, which means they penetrate deeper than small ones;
- droplets of small sizes are quickly decelerated by the gas flow, lose momentum and lag behind large ones;
- as the front of the atomization plume advances, some of the smaller droplets connect with each other, which leads to an increase in d_{32} in measurement localizations more distant from the tip of the nozzle.

Influence of fuel properties on atomization quality. The formation of an aerosol is significantly influenced by the properties of the sprayed liquids themselves and, first of all, their density, viscosity and surface tension forces. In Fig. 5.191 presents dependencies characterizing the process of formation of aerosol droplets as a function of time from the start of injection for light and heavy motor fuel.

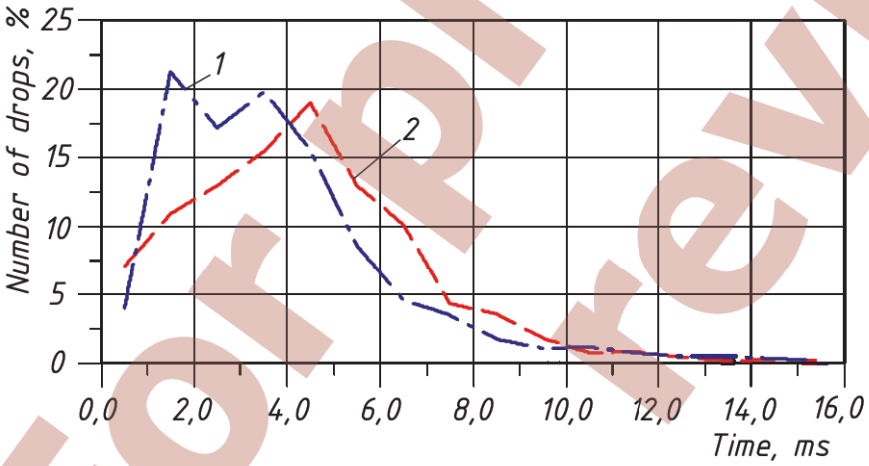


Figure 5.191 – Dependencies characterizing the process of formation of aerosol droplets as a function of time from the start of injection for light and heavy motor fuel: 1 – heavy fuel (viscosity 12 cSt); 2 – light fuel (viscosity 3 cSt) (adapted from [9])

It is clear from the graphs that heavy fuel, other things being equal, forms the bulk of the aerosol at the initial stage of injection, while for light fuel the peak of aerosol formation occurs at 4.3 ms after the start of injection.

This pattern is a consequence of the higher density and viscosity of heavy fuel, as a result of which the disintegration of the atomization plume core begins closer to the tip of the nozzle (Fig. 5.178). This leads to a longer hovering of the droplets, as a result of which some of them connect with each other, which, other things being equal, leads to an increase in the average diameter of droplets of heavy fuel compared to light (Fig. 5.186 *b*).

For preliminary review

The influence of the nozzle hole diameter (d_p) on the quality of atomization. The diameter of the nozzle hole is an important parameter that significantly affects the quality of fuel atomization. The required total flow area of the nozzle hole can be provided by different combinations of diameters and number of nozzle hole in the fuel nozzle tips. As experimental data show, with a decrease in diameter and an increase in the number of hole, other things being equal, the quality of atomization improves.

From Figure 5.186 and it can be seen that as d_p increases, the spraying becomes coarser and less uniform. The taper of the atomization plume increases with increasing diameter of the nozzle holes.

The first circumstance is explained by the fact that, as the diameter of the nozzle holes increases, more time is required for the disintegration of the fuel jet, since the disintegration of the liquid core begins later in time and at a greater distance from the tip of the nozzle. In addition, as d p increases, the role of initial disturbances decreases.

The second circumstance is explained by the fact that as the diameter of the nozzle hole opening increases, the cone angle of the atomization plume decreases, since the length of the fuel atomization plume increases approximately in proportion to the square root of the nozzle hole diameter (5.1, 5.2), and the fuel supply is proportional to the square of the diameter. As a result, the diameter of the atomization plume increases more slowly than its length (Fig. 5.192), and the length of the atomization plume increases.

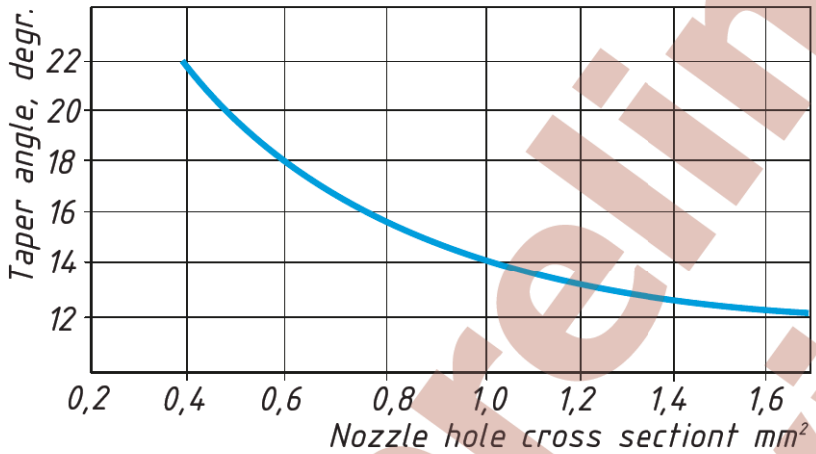


Figure 5.192 – Change in the cone angle of the atomization plume as a function of the diameter of the fuel injector opening of the nozzle (adapted from [9])

In practice, based on the above circumstances, the selection of hole diameters and their number is one of the most significant factors in optimizing mixture formation in diesel engines, since the task of mixture formation is not only to obtain a fine aerosol, but also its uniform distribution throughout the volume of the combustion chamber.

If the diameter of the nozzle holes is too small, the range of the atomization plume decreases, and small drops, having a low impulse, quickly lose their speed and concentrate around the nozzle, where there is not enough air for complete combustion of the fuel. At the same time, air that is not involved in the combustion process remains at the periphery of the chamber. The situation is further aggravated by the fact that an increase in the conicity of the atomization plumes from several peripheral holes leads to the interpenetration of the aerosol of one jet into another (Fig. 5.193 *a*). As a result, zones are formed that are extremely over-enriched in fuel, in which efficient combustion is simply impossible. The appearance of overlapping zones of plumes can also be caused by the mobility of the charge in the combustion chamber, especially when the injectors are off-center. In this regard, when optimizing mixture formation processes, it is necessary to take into account the influence on this process of the degree and nature of turbulization of the air charge in the combustion chamber.

As the nozzle holes diameter increases, the atomization quality deteriorates. But larger drops, having greater momentum, reach the periphery of the combustion chamber. This ensures a more uniform distribution of fuel throughout its entire volume, which leads to more efficient combustion (Fig. 5.193 *b*).

For Preliminary review

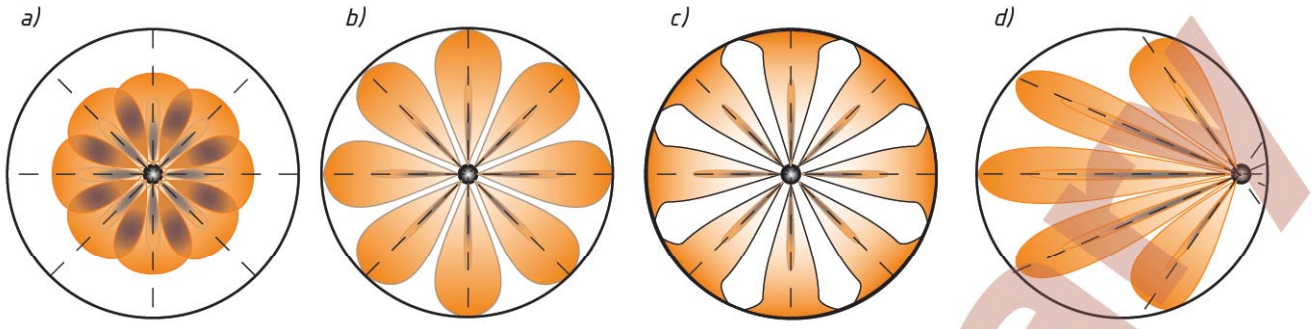


Figure 5.193 – Distribution of atomization plumes depending on the diameters and location of the nozzle holes: *a* – overlap of plumes at increased cone angles; *b* – optimal distribution of plumes throughout the volume of the combustion chamber; *c* – fuel impingement of atomization plumes onto the walls of the cylinder liner; *d* – use of nozzle holes with different diameters with a peripheral location of the fuel injector (adapted from [9])

If the diameter of the nozzle holes is too large, an increase in the atomization plume range can lead to some of the droplets falling onto the relatively cold walls of the cylinder liner, as a result of which the combustion quality will significantly deteriorate (Fig. 5.193 *c*).

The ability to adjust the taper and range of the fuel atomization jet using the hole diameter is especially valuable when fine tuning the mixture formation process in engines with peripherally located injectors. Often, to uniformly cover the volume of the combustion chamber with aerosol, nozzle holes of different diameters are used in the nozzles of such engines (Fig. 5.193 *d*).

Modern research into accumulator injection systems has shown that for high pressures upstream of the fuel injector there is no clear correlation between the average droplet diameter and the fuel injector orifice diameter. This insensitivity is especially characteristic of the region of complete decay of the atomization plume core, which is located quite far from the point of dispersion onset. This pattern is explained by the fact that when spraying through a small hole, aerosol droplets with a smaller diameter can be obtained, however, dispersed disintegration of the atomization plume core also occurs earlier. This leads to an increase in the soaring time of the droplets, during which they combine into larger ones. With a larger hole diameter, dispersion begins later, as a result of which the droplet penetration distance is reduced and the number of connected droplets becomes smaller.

The influence of injection pressure on the quality of fuel atomization. Numerous studies of atomization processes have shown that with an increase in the speed of fuel flow from the fuel injector, the process of internal disturbances in the liquid core intensifies, which has a beneficial effect on the efficiency of its disintegration. The effect of crushing the liquid core from the action of aerodynamic forces also increases. Together, the action of both factors leads to an increase in the quality of atomization. The atomization dispersion required to ensure high-quality combustion of fuel in a diesel engine is achieved at a fuel flow rate from the fuel injector w = 250...400 m/s, which depends primarily on the pressure in front of the nozzle p_{inj}.

As can be seen from table. 5.9 and fig. 5.194, with an increase in injection pressure, the quality of fuel atomization improves, which is expressed in a decrease in d_{32} drops.

For high-quality atomization of heavy fuels, the injection pressure must be sufficiently high (above 60 MPa), which is associated with higher viscosity. In other words, heavy fuels require much more shear energy to achieve the atomization levels of light fuels.

In Fig. 5.194 shows the change in d_{32} depending on the injection pressure obtained for the accumulator system.

It should be noted that at lower injection pressure the d_{32} value has a wider spread. This indicates that under such conditions it is impossible to obtain stable aerosol formation parameters. For injection pressures less than 28 MPa the accumulator system was generally unable to produce aerosol from the heavy fuel.

From Figure 5.194 it is clear that at high injection pressures, the influence of pressure on the fineness of fuel atomization weakens, while the processes of merging small droplets into larger ones

For preview

intensify. Thus, there is a certain limiting pressure at which the droplet sizes reach their minimum, and a further increase in injection pressure does not lead to an improvement in mixture formation.

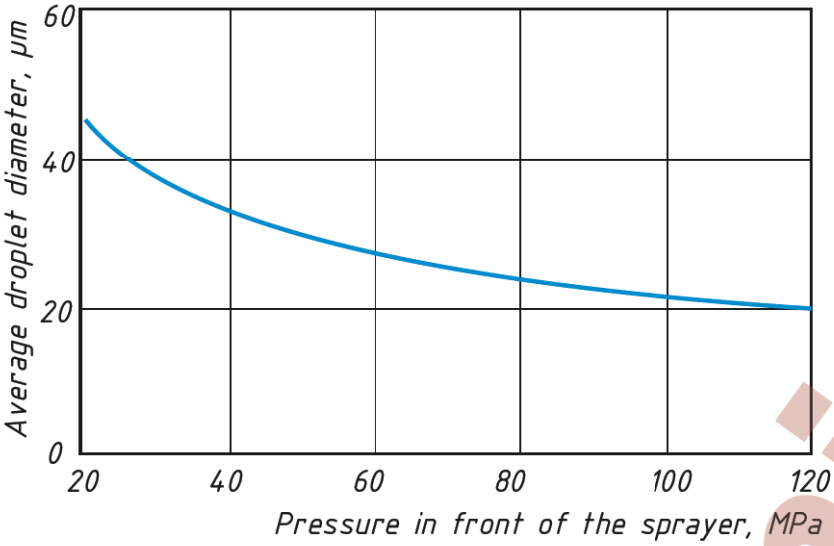


Figure 5.194 – Effect of injection pressure on d_{32} drops of light fuel aerosol (d_p =0.4 mm; ρ_a =25 kg/m^3; s = 120 mm) (adapted from [9])

5.9.5 Hydrodynamic processes in the fuel supply system

To ensure economical and reliable operation of a diesel engine, injection of a cyclic portion of fuel into the combustion chamber must be carried out during a certain angle of rotation of the crank with a given advance angle relative to TDC. In this case, as has already been shown above, the fuel must be sprayed into an aerosol with a given average droplet diameter and given parameters for the distribution of the aerosol over the volume of the combustion chamber.

Fuel injection is a complex hydrodynamic process, the characteristics of which depend on many factors, which include:

- type of fuel system;
- design features of the system elements (type of injection pump and injectors, type of drive mechanism, location and design of the injection valve, length and volume of injection pipelines, etc.);
- technical condition of system elements;
- engine speed, load, etc.

Thus, the assessment of engine performance and the stability of its operating modes under different conditions directly depend on the fuel supply characteristics.

The process of supplying fuel to the combustion chamber of an engine can be divided into two sub-processes: injection of fuel by a pump into the high-pressure line and injection of fuel by an injector. Depending on the type of fuel system, these processes can occur in parallel (direct-acting systems with mechanical and hydraulic drive) or independently of each other (accumulator systems).

Having accepted the assumption that there are no fuel leaks through leaks in the fuel pump and injector, and also if we neglect the viscosity and compressibility of the fuel, the elasticity of the high-pressure line elements, then we can assume that the volumetric flow the fuel pump should be equal to the volumetric fuel flow of the injector (injectors): $G_p = G_i$.

The volumetric second consumption of fuel by an injector (m^3) can be expressed by the continuity equation:

$$G_i = f_n \omega_f, \tag{5.3}$$

where f_n is the total flow area of the nozzle holes (m^2) ($f_n = f_{nh}\, i_{nh}$ where f_{nh}, i_{nh} – area (m^2) and the number of nozzle holes); ω_f – speed of fuel flow from the nozzle holes (m/s).

For conventional cylindrical fuel injectors, the speed of fuel flow from the fuel injector opening (m/s) can be found from the expression:

For preliminary review

$$\omega_f = \mu_n \sqrt{2\frac{p_{inj} - p}{\rho_f}}, \quad (5.4)$$

where μ_n is is the flow coefficient which takes into account the difference in fuel consumption, determined by the Bernoulli equation, and actual consumption; p – gas pressure in the cylinder (Pa); ρ_f – fuel density, (kg/m^3); p_{inj} – fuel pressure in front of the nozzle holes.

With taking into account (5.4), equation (5.3) is transformed to the form:

$$G_i = \mu_n f_n \sqrt{2\frac{p_{inj} - p}{\rho_f}}. \quad (5.5)$$

As for the cyclic fuel supply, it is expressed by the equation (kg/cycle):

$$g_{ci} = \mu_n f_n \tau_i \sqrt{2\rho_f (p_{inj} - p)}, \quad (5.6)$$

where τ_i is the duration of fuel supply by the fuel injector.

As can be seen from equation (5.6), the cyclic fuel supply of the fuel injector is directly related to the pressure in front of its nozzle p_{inj}, which is a dynamic parameter of the injection process.

In accumulator systems, the value of p_{inj} depends on the pressure in the accumulator, its volume and the hydraulic characteristics of the high-pressure line from the accumulator to the nozzle. Accumulators with a large volume allow you to maintain a fairly stable pressure throughout the entire injection, since the value of the cyclic supply is disproportionately small in relation to the volume of the accumulator. When using small-volume accumulators, it discharges during injection, which leads to a decrease in pressure at the end of injection.

In direct-acting systems, the value of p_{inj} depends on the speed of the plunger, wear of the nozzle holes of the nozzle, the density of precision pairs of the pump and fuel injector, the level of residual pressure, etc.

Thus, the cyclic flow of the injector g_{ci} will be less than the geometric flow of the pump g_{gp}, therefore it is also called the actual cyclic flow.

In accumulator systems, the amount of cyclic flow is determined by the opening time of the control valve τ_i and is set by the microprocessor engine control unit.

The volumetric flow of the fuel pump in accumulator systems must compensate for the fuel consumption for injection and can be defined as the simple displacement of a measured portion of fuel into the accumulator by a plunger:

$$G_v = f_p h_a k, \quad (5.7)$$

where f_p – area of the plunger end (m^2); h_a – the magnitude of the active stroke of the plunger (m); k is the proportionality coefficient between the injection flow rate and the pump supply, taking into account the fuel consumption for controlling the injector.

The value of the coefficient k depends on the number of pump strokes per revolution of the pump shaft, the number of pumping sections and their performance, and the design of the fuel injector, which determines the amount of fuel consumption for control. As a rule, in most designs the performance level can be adjusted, which allows you to change the volumetric fuel supply to the accumulator depending on the fuel consumption for injection.

For fuel systems with a hydraulic plunger drive, the value of τ_i depends on the average speed of the plunger, and for systems with a mechanical drive, on the average speed of the plunger in the active stroke section.

Taking this into account, the volumetric flow of the fuel pump (m^3/s) is equal to:

$$G_p = f_p c_p, \quad (5.8)$$

where f_p is the cross-sectional area of the plunger (m^2); c_p – plunger speed (m/s).

Plunger speed (m/s):

For preliminary review

$$c_p = \frac{dh_p}{d\tau} = \frac{dh_p}{d\varphi}\frac{d\varphi}{d\tau}, \quad (5.9)$$

where $dh_p/d\varphi$ – speed of lifting the plunger along the angle of rotation of the crankshaft (m/s); $d\varphi/d\tau$ – angular velocity (rad/s).

At constant speed n angular velocity (rad/s)

$$\omega = \frac{d\varphi}{d\tau} = \frac{\pi n}{30}. \quad (5.10)$$

Taking into account expressions (5.9) and (5.10), equation (5.8) of the volumetric flow of the fuel pump (m^3/s) takes the form:

$$G_p = f_p \frac{dh_p}{d\varphi}\frac{\pi n}{30}. \quad (5.11)$$

Equating the right-hand sides of equations (5.5) and (5.11) based on the initial position that the volumetric fuel flow rate of the fuel injector is equal to the volumetric flow of the pump ($G_i = G_p$) and, solving the equality with respect to p_{inj} (Pa), we obtain:

$$p_{inj} = p + \frac{\rho_f \pi^2}{1800}\left(\frac{f_p}{\mu_n f_n}\right)^2\left(\frac{dh_p}{d\varphi}\right)^2 n^2. \quad (5.12)$$

Analysis of equation (5.12) shows that the injection pressure depends on the density of the fuel, the ratio of the area of the fuel pump plunger to the effective flow area of the fuel injectors $f_p/(\mu_c f_c)$ the plunger lifting speed at the considered moment of injection $dh_p/d\varphi$ and rotation speed p diesel.

Due to the assumptions made, equation (5.12) allows us to estimate the nature of the influence of the main factors on the injection pressure with a certain degree of accuracy. In real conditions, at pressures typical of marine diesel fuel systems, liquid fuel behaves like an elastic medium that contracts under the influence of force and expands when this force is removed.

If a disturbance is created in the volume of liquid, it will propagate in the form of a pressure wave at the speed of sound (for fuel on average 1450 m/s). When the wave encounters an obstacle, it is partially or completely reflected from it, causing a reverse pressure wave to appear. As a result of the disturbances created by the plunger, direct pressure waves occur in the pipeline. Reflecting from obstacles, which can be shut-off valves, injector needle valves, fuel injector openings, etc., reverse pressure waves arise in the high-pressure line. Overlapping each other, pressure waves create a complex picture of processes, the nature of which largely determines the efficiency of fuel atomization. In particular, under the influence of pressure waves arising from the impact of the mass of fuel on the closed needle valve of the injector or the reverse wave from the closed injection valve of the injection pump, so-called after-injections can be observed – additional lifts of the needle after the end of the main injection. During the injection period, the fuel pressure slightly exceeds the pressure at which the injector starts feeding, so the atomization quality is very low. The fuel, flowing out of the nozzle holes, partially settles on the tip of the nozzle, causing severe carbon deposits. In addition, fuel injection occurs, as a rule, on the expansion line, which leads to an increase in the afterburning period and a decrease in diesel efficiency.

The occurrence of after-injections can be either a consequence of incomplete refinement of fuel equipment at the factory, or the result of changes in technical condition during operation. The occurrence of after-injections is most likely in a fuel supply system with a long discharge pipeline; it is for this reason that they try to place fuel pumps as close to the injectors as possible

When designing fuel equipment, complex hydrodynamic calculation methods are used that take into account the viscosity of the fuel and its compressibility, the volume and length of the injection pipelines, the presence of fuel leaks in the fuel pump and injector, the influence of direct and reflected waves on the nature of pressure changes in various sections. Three-dimensional computer

For preliminary review

modelling methods are being widely introduced into design practice, allowing not only to obtain numerical solutions, but also to visualize the processes occurring in fuel systems.

5.9.6 Injection pressure characteristic

From the point of view of an ideal atomization process, it is desirable that the pressure in front of the fuel injector remains constant throughout the entire injection process or has a maximum at the beginning of injection, when the first portions of fuel are introduced into the cylinder, ensuring self-ignition. However, in real processes, the pressure at which fuel is injected into the cylinder through a fuel injector is not constant and the nature of its change is, as a rule, far from ideal.

The nature of the change in pressure in front of the fuel injector largely depends on the type of fuel system, engine operating mode, condition of the fuel equipment elements and a number of other factors. In Fig. 5.195 shows the dependence of the change in pressure in front of the fuel injector with respect to the rotation angle, called the *injection pressure characteristics* for the three main types of fuel systems used in modern MICEs.

The most stable pressure throughout the entire injection is ensured by the low-speed diesel accumulator system of the RT-flex series from Wärtsilä. The presence of a large volume of accumulating space allows the pressure to be maintained at a constant, fairly high level throughout the entire injection, regardless of the engine operating mode.

Stable injection is ensured by a fuel supply system with a hydraulic, electrically controlled injection pump drive, used on MAN ME series engines. The presence of a hydraulic drive allows us to obtain a law of fuel supply to the combustion chamber that is practically independent of the engine speed.

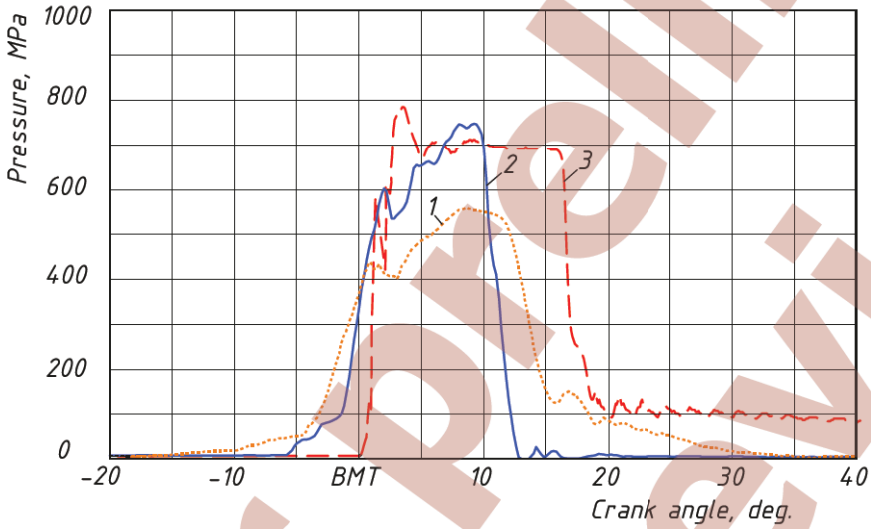


Figure 5.195 – Characteristics of changes in pressure in front of the injector nozzle for various fuel injection systems at 75% of the rated power: 1 – direct-acting system with a mechanical drive of the fuel pump plunger (diesel S50MC-C); 2 – direct action system with electro-hydraulic drive of the fuel pump plunger (diesel S50ME-C); 3 – accumulator injection system (diesel RT-flex50B) (adapted from [115])

For diesel engines of the MC series of the same company, equipped with a mechanically driven injection system, as the rotation speed decreases, a decrease in injection pressure is observed, proportional to the decrease in the plunger speed.

Fuel supply parameters that determine the nature of the injection process are divided into static (geometric) and dynamic.

Static parameters characterize the process of fuel supply by a high-pressure pump, dynamic parameters – by an injector. These parameters characterize the fuel supply from a qualitative point of view, they show how the fuel injection phases are located relative to TDC of the piston and determine: the beginning, end and duration of fuel supply by the pump (φ_{sfsp}, φ_{efsp}, φ_p) and the fuel injector (φ_{sfsi}, φ_{efsi}, φ_i). These data are the basis for analyzing combustion processes, economic and dynamic indicators of the engine's operating process.

The mutual influence of the static and dynamic phases of fuel supply is shown in Fig. 5.196. It shows that the dynamic phases are shifted relative to the static ones in the direction of rotation of the crankshaft. The main reason for this phase shift is the elasticity of the fuel filling the high-pressure line and the elastic deformation of the high-pressure fuel pipes.

For preliminary review

Shown schematically in Fig. 5.196 pressure change curves in the cavity of the fuel pump (p_p) and in front of the fuel injector (p_{inj}) are typical for direct-acting systems with a delivery valve installed in the pump. In addition to the pressure curves, the diagram shows a graph of injector needle rise h and a pie chart of the fuel supply process.

Before delivery begins, the working cavity of the pump is filled with fuel under pressure p_{sfsp} created by the booster pump. After the upper edge of the plunger covers the filling hole (φ_{sfsp}), a sharp increase in pressure p_p is observed, which indicates the beginning of *the active stroke of the plunger*. The angular gap between the start of fuel supply by the pump and TDC of the engine determines *the supply advance angle by pump* (φ_{pdt}).

The delivery valve installed in the fuel injection pump opens when the pressure p_p increases to the residual pressure p_{rp} maintained in the discharge line between injections. Until this moment, the injection system is blocked on one side by the fuel injector needle, and on the other by the pump delivery valve.

After opening the delivery valve, pressure will increase along the entire discharge line. The pressure wave created by the plunger moves towards the fuel injector and, upon reaching it, leads to an increase in pressure in front of the fuel injector p_{inj}. Upon reaching p_{inj} injector start pressure p_{sfsi}, the value of which is determined by pre- tightening the needle valve spring, the needle rises, passing fuel into the nozzle holes.

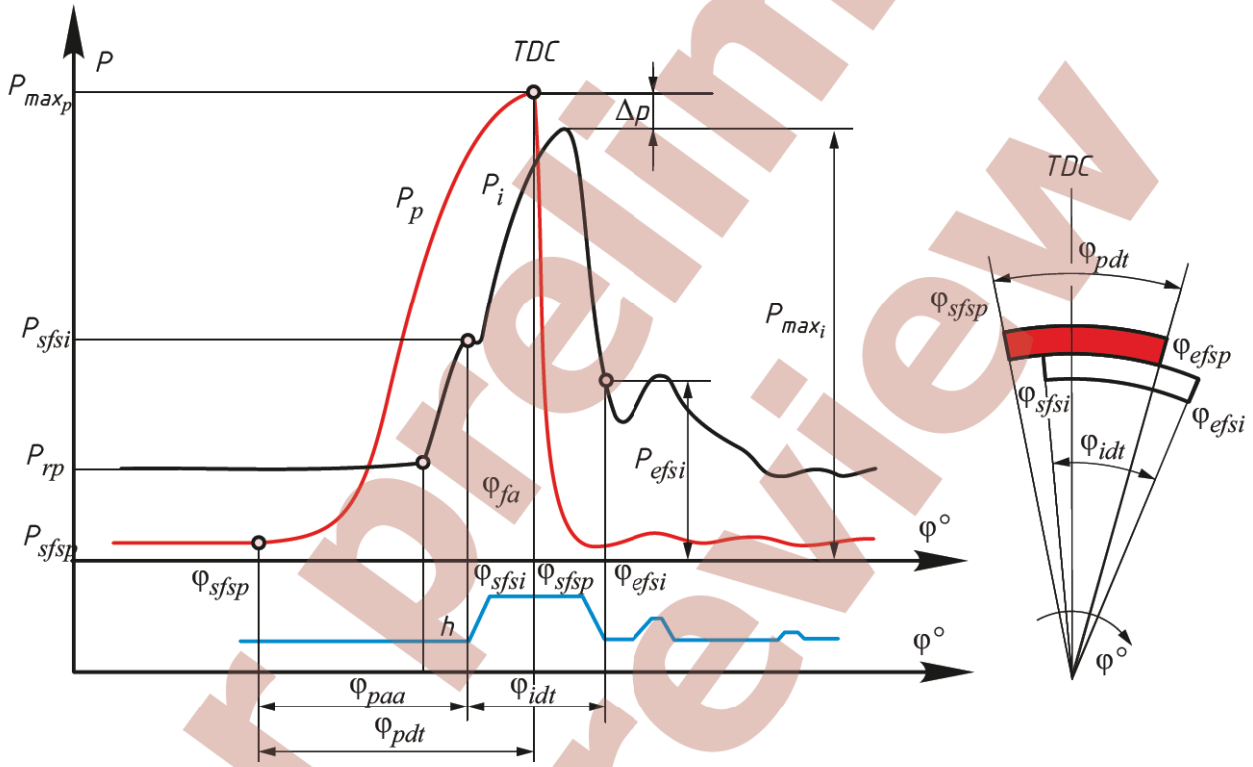


Figure 5.196 – General patterns of pressure changes in the cavity of the fuel pump (p_p) and in front of the fuel injector (p_{inj}) for direct-acting fuel supply systems, combined with a diagram of the rise of the injector needle and a pie diagram of the fuel supply process (adapted from [9])

The moment of appearance of a jet of fuel from the nozzle holes of the fuel injector, related to the position of the engine crankshaft, is called the *angle injection start* (φ_{sfsi}). The beginning of pump delivery, related to the position of the piston at TDC, is called *the static advance angle of* fuel supply (φ_{pdt}).

The delay in the start of the injector supply relative to the start of the pump supply is determined mainly by the time required to increase the fuel pressure in the injection system from the pumping pressure ($p_{bp} = p_{sfsp}$) to the pressure at the start of the lifting the needle of the sprayer (p_{sfsi}). Therefore, the larger the volume of the system, the lower the residual pressure p_{rp} and the stronger the spring tension, the greater the delays angle of the fuel injector supply φ_{ppa}. The angular gap between

For preliminary review

the start of the injector supply and the engine TDC is called *the dynamic advance angle of the injector supply* (φ_{fa}). If injection is carried out before the piston reaches TDC, the advance angle has a positive value; if after, it is negative.

As can be seen from Figure 5.196, at the moment the fuel injector opens, a characteristic dip is noted on the p_{inj} curve, due to the fact that when the needle is raised, the volume of the sub-needle space increases. After placing the needle on the stop, the increase in the sub-needle space stops and the increase in pressure continues. A characteristic dip is also present in the diagrams shown In Fig. 5.195.

The coincidence of the cut-off edge with the discharge hole (or the opening of the shut-off valve) is accompanied by a sharp drop in pressure p_p. The delivery valve closes, and the fuel, under the influence of a pressure difference, is quickly transferred into the low-pressure cavity. This moment corresponds *to the end of pump delivery* (φ_{efsp}). The angular interval between the beginning and end of the supply is called *the duration of the pump supply* φ_{pdt}.

After some time, a wave of pressure drop p_p from the pump reaches the fuel injector and further injection occurs only due to the expansion of the fuel, causing the pressure p_{inj} falls. When it drops to the pressure level p_{efsi}, the spray needle sits on the seat (φ_{efsi}). The angular gap between the beginning and end of the fuel supply by the injector is called *the duration of the injector supply* φ_{idt}.

From rice. 5.196 it can be seen that the pressure p_{inj} at the end of the fuel supply by the fuel injector is less than at the moment the supply begins. This phenomenon is called *the differential needle effect.* It is explained by the fact that at the moment the fuel injector opens, the pressure in the fuel injector cavity acts only on the part of the end surface of the needle valve that is not pressed against the seat, creating less force than when the needle is open and the pressure acts on its entire end surface.

When regulating the fuel pump at the beginning of delivery with a decrease in engine load, the delivery angle increasingly shifts to the area where the plunger speed decreases. This leads to an imbalance between the pump and injector flows. When the plunger speed decreases at low speeds, the pump supply becomes so sluggish that the fuel injector needle settles into place before the active stroke of the plunger ends. It is for this reason that this method of regulating the cyclic supply in its pure form is practically not used on marine diesel engines.

When regulating the injection pump at the end of supply, at the moment of cut-off, the pres-sure drops sharply and a reverse flow of fuel is formed through the still raised check valve located in the pump. The closing of the valve is accompanied by a hydraulic shock, which generates pressure waves travelling to the fuel injector. Due to the energy of these waves, the injector needle can continue to stand on the stop, delaying the injection the longer the greater the cyclic supply. If the needle valve has already closed by the time the wave arrives, the wave can open it again. If the pressure at the wave front exceeds the opening pressure of the fuel injector, after-injection will occur.

From Figure 5.196 it is clear that as the plunger rises, the pressure in front of the nozzle first increases from the pressure at the beginning of the fuel injector supply p_{sfsi} to a certain maximum $p_{\max inj}$, and then drops to the pressure at the end of the fuel injector supply p_{efsi}.

Taking into account the variable nature of the pressure in the system, the term *injection pressure* usually means *the maximum pressure* in front of the nozzle $p_{\max inj}$. It is the level $p_{\max inj}$ that determines the hydraulic loads on the elements of the high-pressure lines and their connections, i.e., ultimately, the reliability of the fuel equipment.

Over the past few years, there has been a steady trend of increasing injection pressure in order to shorten the injection period and improve the quality of fuel atomization, which ultimately improves the efficiency of diesel engines. In modern marine diesel engines, the injection pressure ranges from 60...200 MPa, and in some cases can reach up to 250 MPa.

The injection duration is determined by the moments of lifting and landing of the fuel injector needle (line *h*, Fig. 4.196). For marine diesel engines it is φ_i =20...40°.

As was shown above (5.12), the nature of the change in injection pressure depends on the design and operational parameters of the elements of the fuel supply system, their operating mode and the physical properties of the fuel.

For preliminary review

The design parameters, first of all, include the lifting speed of the plunger $c = dh_p/d\varphi$, which for systems with a hydraulic drive depends on the speed of entry of control oil into the cavity of the hydraulic cylinder, and for systems with a mechanical drive – on the profile of the cam washer of the fuel pump.

With a mechanical drive of the injection pump, the choice of the fuel cam profile is carried out on the basis of calculations of the main geometric dimensions of the fuel equipment and the kinematic characteristics of the fuel pump plunger (the average speed of the plunger c_m on the interval of the geometric useful stroke of the plunger).

The main parameters include those that provide the specified characteristics of fuel injection over the duration of supply: the geometric dimensions of the working profile of the fuel cam, the lift angle and the amount of full rise of the profile.

In marine diesel engines, fuel cam profiles are most often used, defining trapezoidal and triangular or similar laws for changing the speed of the plunger depending on the angle of rotation of the cam shaft (Fig. 5.197).

The first of these profiles (Fig. 5.197 *a*) is characterized by a constant plunger speed during fuel injection, which creates certain convenience when regulating the fuel equipment on the engine according to injection advance. The second (Fig. 5.197 *b*) allows you to obtain the highest average speed of the plunger in the section of its active stroke and make the most of the given full rise of the fuel cam profile.

Quite often there are cases when only a section of the ascending velocity branch is used as a working one (Fig. 5.197, *c*).

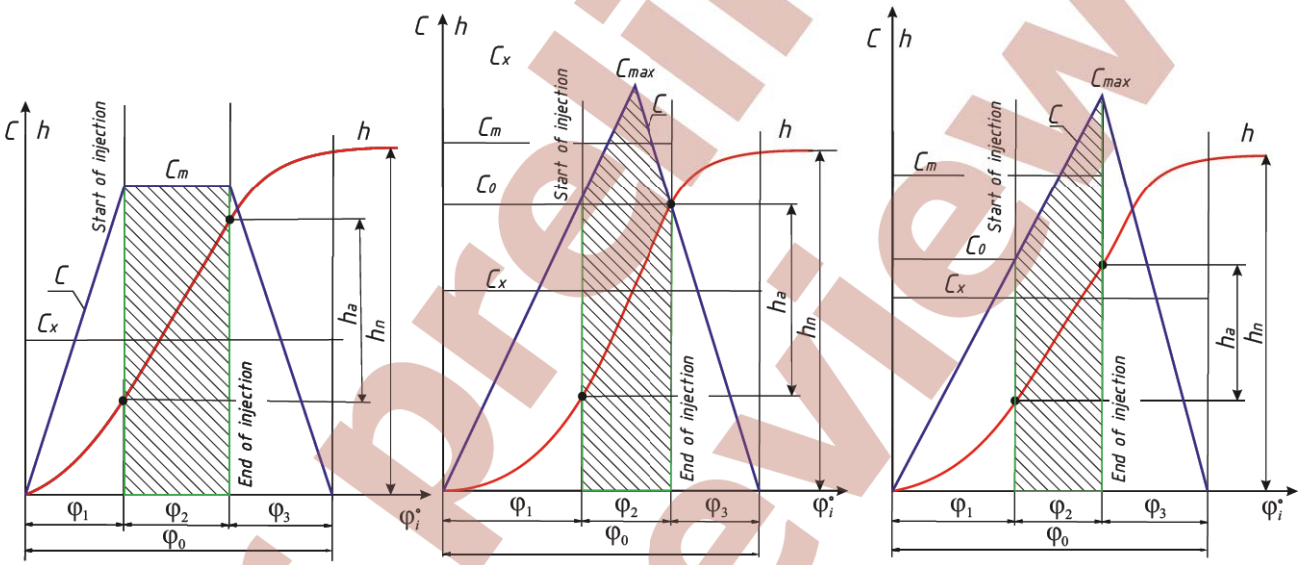


Figure 5.197 – Typical characteristics of fuel cam profiles: *a* – trapezoidal law of change in plunger speed; *b* – triangular law with equal plunger speeds at the beginning and end of injection; *c* – triangular law using the section of the ascending velocity branch; *c* – plunger speed; *h* – plunger stroke; φ – angle of rotation of the cam shaft; c_{max} – maximum plunger speed; c_m – average speed in the section of the active stroke of the plunger h_a; c_0 – speed corresponding to the beginning of the geometric active stroke of the plunger (start of supply); c_x – average speed over the period of full stroke of the plunger h_p; h_p – full stroke of the plunger (raising the profile of the fuel cam); h_a – active stroke of the plunger; φ_0 – fuel cam profile elevation angle; φ_1 – duration of the plunger lifting phase until the geometric start of fuel supply; φ_2 – geometric injection duration; φ_3 – duration of the fuel cut-off phase (adapted from [9])

The average plunger speeds for fuel cam profiles with a triangular law of speed change, all other things being equal, are 6...12% higher.

The permissible acceleration of the plunger usually lies in the range of 200...400 m/s^2, and in some cases it can reach 500 m/s^2 or more.

The acceleration value is the starting point for choosing a plunger spring, which must ensure constant contact of the pusher roller with the fuel cam profile.

For preliminary review

In practice, quite often, to ensure the specified injection parameters, asymmetric laws for changing the speed of the plunger are used, in which the highest speed is achieved in the section when the plunger is lifted by the middle, steepest section of the cam washer profile. During this period, the injection pressure reaches its maximum, ensuring high atomization quality.

Figure 5.198 shows diagrams of the speed and displacement of the injection pump plunger for the case when the period of fuel supply by the pump (geometric supply period) φ_2 falls on a section of high values of the plunger speed c_p. Cut-off at high speed of the plunger at the end of the supply ensures a sharp drop in injection pressure and a sharp landing of the fuel injector needle. The injection period at low values of p_{inj} before needle landing is short.

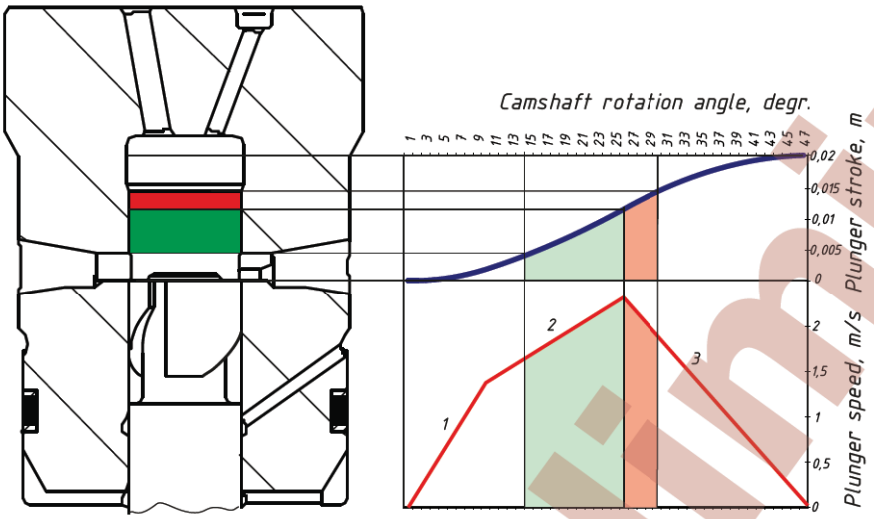


Figure 5.198 – Diagrams of movement and speed of the fuel pump plunger with regulation of fuel supply at the end of delivery: 1 – movement of the plunger before the start of delivery; 2 – area of cyclic supply in the operating load range (green area); 3 – pump supply reserve area (red area) and plunger deceleration area

When operating diesel engines, to set up direct-acting fuel equipment with a mechanical drive, static phases of the fuel supply of the pump, which are also called *geometric*, *are used*. These phases are available for monitoring and regulation without the use of special equipment. In order to ensure the specified actual phases of fuel injection by a fuel injector, it is necessary to establish such geometric phases of fuel supply that would take into account the hydrodynamic properties of the fuel supply system. For ease of analysis, the fuel supply process is divided into separate periods. Based on the above, three such periods can be distinguished (Fig. 5.196):

- *injection delay period* (φ_{paa}) – the angular interval between the start of supply by the pump and the beginning of supply by the fuel injector, caused by the compressibility of the fuel, the elasticity of the discharge pipeline, the final speed of propagation of the pressure wave in it, the residual pressure in the pipeline p_{rp} and needle opening pressure p_{sfsi}. According to experimental data, for marine low-speed diesel engines the duration of the period is 2...20°;

- *active injection period* – the angular interval between the start of supply by the injector (φ_{sfsi}) to the end of supply by the pump (φ_{efsp}), during which the main part of the cyclic portion of fuel is injected into the cylinder. Its duration depends on the diesel load. Pattern of pressure changes during the period active injection largely depends on the lifting speed of the injection pump plunger;

- *free flow period* – the angular interval from the end of the pump supply (φ_{efsp}) to the end of the fuel injector supply (φ_{efsi}). The injection process occurs due to the energy of the compressed fuel and the elasticity of the injection pipeline. Fuel injection occurs at a gradually decreasing pressure p_{inj}, which causes a deterioration in the quality of atomization. During this period, fuel is injected at the expansion line in the cylinder, which leads to an increase in the duration of fuel burnout and a decrease in diesel efficiency.

5.9.7 Fuel injection characteristics

The main goal of optimizing the parameters of the injection process is to obtain a given *injection characteristic* or *supply law*, which is the dependence of the mass (volume) of fuel g_φ passing through the fuel injector on time or the angle of rotation of the camshaft φ (Fig. 5.199 *a*). In other

For preliminary review

words, the supply law shows how the cyclic portion of fuel supplied during the injection period is distributed for each degree of crank rotation.

In practice, the *integral* characteristic of the injection process is often used (Fig. 5.199 *b*), which represents the total fraction of the cyclic portion $G_{\varphi i}$ injected into the cylinder from the start of injection to the moment under consideration, depicted graphically as a function of the crank angle φ or time.

The injection characteristics and the dynamic factor largely determine the rate of increase in gas pressure in the cylinder during fuel combustion, the dynamism and rigidity of the operating process. The latter have a significant impact on the mechanical stress of parts of the cylinder-piston group and crankshaft, as well as on the wear rate of diesel bearings.

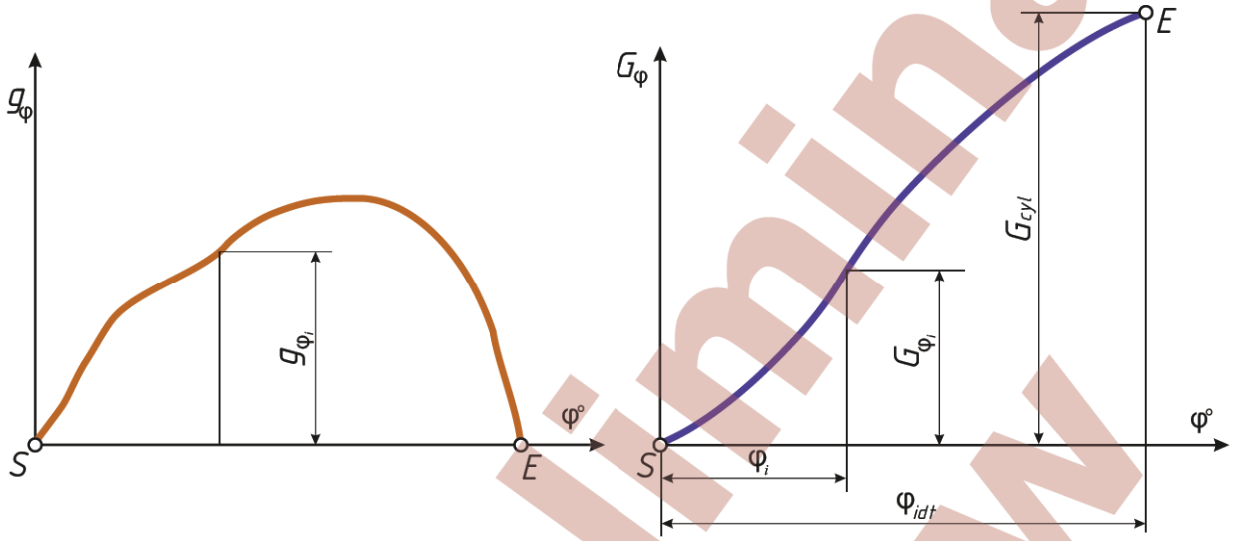


Figure 5.199 – Differential (*a*) and integral (*b*) characteristics of fuel injection: *S* – start of supply; *E* – end of supply (adapted from [9])

5.9.8 Fuel combustion in diesel engines

Fuel combustion in the engine working space is a key stage of the operating process, the nature of which determines the efficiency of the engine, its environmental performance and reliability. In its most general form, combustion refers to rapidly occurring chemical reactions of fuel oxidation, accompanied by the release of heat and the appearance of a flame. During combustion, the chemical energy of the fuel is converted into thermal energy of combustion products, which is converted into mechanical work during the expansion stroke. In a real operating process, from the beginning of injection to the end of burnout of a cyclic portion of fuel, complex physical and chemical sequential-parallel processes of atomization, evaporation, ignition and combustion of fuel occur. These processes overlap and significantly influence each other, which complicates their modeling and forecasting. For example, injection occurs in parallel with the heating and evaporation of fuel droplets previously supplied to the combustion chamber. The onset of combustion in the working cylinder leads to an increase in pressure and temperature, which accelerates the heating and evaporation of fuel droplets previously entering the engine combustion chamber, etc.

To analyze the indicator diagrams of the engine operating process in the fuel supply and combustion part, a pressure curve is used, which can be obtained directly on the engine and a temperature change curve obtained analytically from the pressure curve (Fig. 5.200). There are four periods (phases) that differ in the nature of the physical and chemical processes:

- auto-ignition delay period (section 1-2);
- period of fast (kinetic) combustion (section 2-3);
- period of slow (diffusion) combustion (section 3-4);
- period of afterburning (section 4-).

It should be noted that such a division is very conditional; there are no clear boundaries between the end of one period and the beginning of another.

For preliminary review

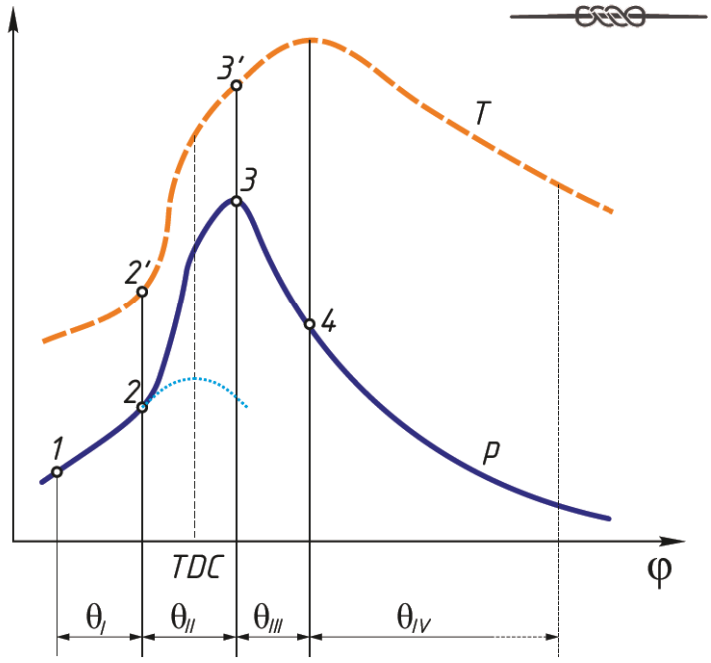


Figure 5.200 – Change in temperature (T) and pressure (p) during fuel combustion as a function of the crankshaft rotation angle (φ)

The auto-ignition delay period (τ) covers part of the cycle (θ_I) from the beginning of fuel entering the cylinder (point 1 Fig. 5.200) until the moment the pressure curve separates from the compression curve (point 2 Fig. 5.200), which is conventionally taken as the beginning of fuel combustion. During this period, complex physical and chemical processes occur in the cylinder: the spread of a spray of atomized fuel in the volume of the combustion chamber, heating and evaporation of droplets, the breakdown of fuel molecules into radicals and atoms under the influence of high temperature, the oxidation of combustible elements by atmospheric oxygen.

Period of rapid (kinetic) combustion (θ_{II}) covers part of the cycle from the beginning of self-ignition of the fuel (point 2 Fig. 5.200) until the moment the maximum pressure is reached in the cylinder (point 3 Fig. 5.200). This period is characterized by direct (kinetic) reactions between oxygen in the air and the combustible components of the fuel injected and evaporated mainly during the autoignition delay period τ. Fuel combustion is uncontrollable, occurs with a high rate of heat release and a sharp increase in pressure in the cylinder with continued fuel supply. The intensity of the increase in pressure during this period determines the dynamism (rigidity) of the operating process, on which the loads on the main structural elements of the engine and, ultimately, its reliability depend.

Period of main (diffusion) combustion (θ_{III}) corresponds to the section of the cycle from the moment the maximum pressure p_z is reached in the cylinder (point 3 in Fig. 5.200) until the maximum temperature T_{max} is reached (point 4 in Fig. 5.200). It is characterized by the fact that the combustion process spreads throughout the entire volume of the combustion chamber. The injected fuel enters an open flame environment with high pressure and temperature, which significantly reduces the time for pre-flame processes, due to which it burns almost immediately after entering the working space. As the fuel burns out, the oxygen concentration in the working space decreases, which leads to a gradual predominance of diffusion reactions over kinetic ones and an increase in the concentration of products of incomplete combustion of fuel in the cylinder. The change in pressure in this part of the cycle is not as intense as in the previous period, since the piston begins to move downward, increasing the volume. Thus, the change in pressure depends on the relationship between the characteristics of injection, combustion and the increase in cylinder volume.

In engines operating according to the Miller cycle, most of the combustion process is transferred to the expansion line and, therefore, the main part of the fuel is burned during this period, which makes it possible to limit the maximum cycle pressure p_z by increasing the volume of the engine working space in this section of the cycle.

Fuel burnout period (θ_{IV}) covers the cycle interval between the moments of reaching the maximum temperature (point 4 in Fig. 5.200) in the cylinder and the end of fuel combustion. The moment of completion of fuel combustion is determined by the angle of rotation of the crank at which the rate of heat release $dQ/d\varphi = 0$. This period is characterized by the absence of fuel supply, a de-

For preliminary review

crease in pressure in the cylinder, a further decrease in the oxygen concentration and an increase in the concentration of the final combustion products. For some high-speed diesel engines, this period may last until the exhaust valve opens, which is an extremely undesirable phenomenon. It leads to an increase in heat losses and an increase in the thermal stress of the parts of the cylinder-piston group, and, consequently, to a deterioration in the efficiency of the diesel engine.

In new Miller cycle engines, in order to limit the maximum cycle pressure p_z, most of the combustion process is transferred to the expansion line. As a result, the pressure and temperature of the gases at the end of expansion are higher in these engines. Losses associated with higher gas parameters at the end of expansion in the piston part are compensated by their subsequent expansion in a highly efficient turbocharger unit. This improves the performance of the charging system, which is a key requirement for an efficient Miller cycle.

To analyze the nature of the combustion process, a number of indicators and characteristics are used, which will be discussed below.

Fuel combustion is the main process of the operating cycle, during which heat is released, which is converted in the engine into useful mechanical work. When assessing combustion processes, intermediate physicochemical changes in the working fluid are not considered, but only the final results of chemical reactions of combustible fuel components with atmospheric oxygen are taken into account.

When assessing the parameters of the working fluid during the combustion process, the actual nature of the change in pressure in the cylinder is conditionally replaced by isochoric *cz* and isobaric *zz′* by heat supply processes (Fig. 5.201). This cycle is called a mixed heat input cycle or mixed cycle.

From the moment fuel begins to be supplied to the cylinder (point *h*, Fig. 5.201) until the end of combustion, complex physical and chemical processes of evaporation, ignition and combustion of fuel, heat release, changes in pressure and temperature occur. All these processes occur at finite speeds, which determine the duration of the process as a whole. For most engines, the combustion process occupies a cycle section of 20...35°.

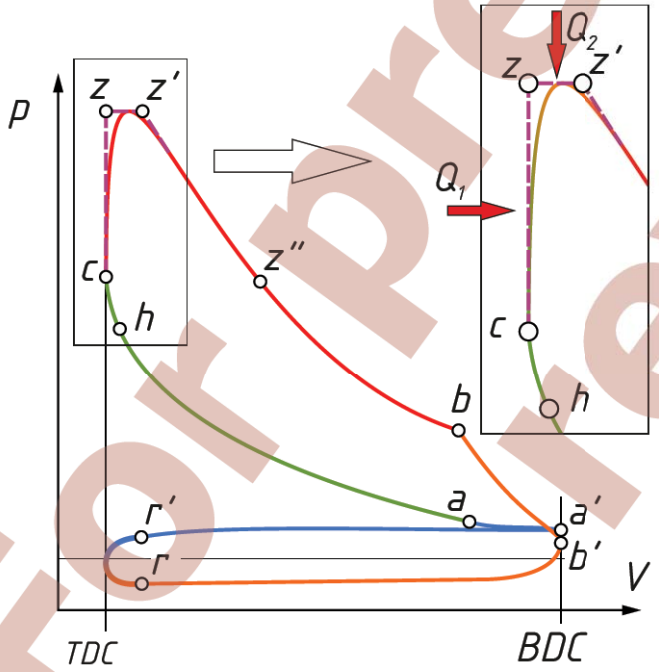


Figure 5.201 – Scheme of the combustion process in a cycle with mixed heat supply

Fuel injection begins a few degrees before TDC (point *h*, in Fig. 5.201). This makes it possible to distribute the fuel aerosol throughout the combustion chamber and ensure the occurrence of physico-chemical processes preceding combustion (pre-flame processes) before the piston reaches TDC.

The distribution of fuel droplets in the air charge environment ensures their rapid heating and evaporation, as well as direct contact of fuel vapor with air oxygen, which acts as an oxidizer.

Atomized fuel mixed with hot air (in area *hc*, fig. 5.202), ignites and burns, as a result of which the temperature and pressure of the gases sharply increase (point *z*), and then their expansion be-

For preliminary review

gins. The piston begins to move from TDC to BDC, making a power stroke. Gases, acting on the piston during the power stroke, perform useful work, which is transmitted to energy consumers through the crank mechanism. The combustion process continues in the *zz′* section, as a result of which the pressure after the piston passes TDC remains relatively constant for some time. Conventionally, point *z′* is considered the end of combustion, although in practice, fuel combustion can continue further, along the expansion line (point *z″*).

The section of conditional combustion *czz′* on the indicator diagram (Fig. 5.202) is called the period *of visible combustion.* In accordance with the first law of thermodynamics, the heat released during fuel combustion is used to increase the internal energy of the working fluid and to perform external mechanical work.

In a mixed calculation cycle, heat Q_1 supplied at a constant volume (section *cz*) is spent on increasing the internal energy of the working fluid, which is accompanied by a sharp increase in pressure and temperature, and heat Q_2 supplied at constant pressure is spent on performing work (section *zz′*). The ratio of these heat shares affects the parameters at the end of visible combustion: maximum combustion pressure $p_{z'}$, combustion temperature $T_{z'}$ and the performance of the cycle as a whole.

If the pressure is known p_c, temperature T_c, mass M_c and volume V_c of the working fluid at the beginning of the combustion process (point *c*), then when determining the main parameters of the working fluid at the end of the «visible» combustion of fuel at point *z′*: maximum combustion pressure $p_{z'}$, temperature of the working fluid $T_{z'}$, mass $M_{z'}$, volume of the cylinder at the end of visible combustion $V_{z'}$, the only known parameter is the volume $V_z = V_s$, and $M_{z'} = M_s + \Delta M$, where ΔM, kmol is the increment in the mass of the working fluid due to fuel combustion.

If the value of the maximum combustion pressure is known based on the engine indication results $p_z = p_{z'}$ the number of unknown parameters is reduced to two – $T_{z'}$ and $V_{z'}$.

Coefficients of heat release and use. The total amount of heat supplied as a result of fuel combustion $Q = Q_1 + Q_2$ is usually determined by the lower calorific value of fuel Q_l, i.e. $Q = Q_l$. In a real engine, fuel combustion is accompanied by inevitable heat losses: *incomplete combustion* Q_{ci}, *dissociation of combustion products* Q_{dis} and *into the water-cooled walls of the workspace* Q_w.

The incompleteness of combustion Q_{ci} is due to the fact that part of the supplied fuel does not have time to burn in the section *czz′*. This is especially true for fuel that has fallen on the cold walls of the cylinder or is located in the near-wall areas, where conditions for combustion are most unfavorable. In addition, the cylinder charge contains a certain amount of incomplete combustion products from the previous cycle.

At temperatures above 2000 K, dissociation of combustion products occurs, i.e. splitting of molecules of certain compounds ($CO_2 \rightarrow C + O_2$; $H_2O \rightarrow H_2 + O$). This process is accompanied by the absorption of heat and a decrease in the temperature at the end of combustion.

An indicator for assessing the amount of heat released during combustion is *the heat release coefficient x*, which is the fraction of the heat of combustion Q_l released at the considered moment of the operating process, and takes into account losses from incomplete combustion Q_{ci} and losses due to *the dissociation of combustion products* Q_{dis},

$$x = [Q_l - (Q_{ci} + Q_{dis})]/Q_l .$$

At the beginning of the combustion process (point *c*) $x = 0$, and at the end of the fuel burnout on the expansion curve (point *z″*) $x = 1$. The change in the heat release coefficient *x* shown in Fig. 5.202 line *0 – 1′– 2′– 3′*, depends on the nature and efficiency of the ignition and combustion processes.

During combustion and further expansion, heat is also lost through the walls of the working space into its cooling medium Q_w. The nature of the change in these losses is shown in Fig. 5.202, line *0′–3–4*.

The share of supplied heat that is used to increase the internal energy of the working fluid and perform external mechanical work on the combustion and expansion lines is estimated by the *heat utilization coefficient* ξ, which takes into account additional heat losses into the cooling medium Q_w

For preliminary review

$$\xi = [Q_l - (Q_{ci} + Q_{dis} + Q_w)]/Q_l = x - Q_w / Q_l ,$$

which takes into account all heat losses during fuel combustion. Therefore, for an arbitrary point in the combustion-expansion processes, we can write $\xi Q_l = xQ_l - Q_w$. Thus, for an arbitrary point in the combustion-expansion processes, the curve *0 – 1 – 2 – 3 – 4* (Fig. 5. 202) of the heat utilization coefficient can be obtained by subtracting losses for heat transfer into the cooling medium Q_w from the heat release coefficient curve. The heat utilization coefficient ξ_{max} reaches its maximum value when the amount of heat released from the combustion of fuel is completely compensated by heat transfer into the walls (point 2). With further expansion of gases, the heat transfer exceeds the amount of heat released during fuel combustion, which leads to a decrease in the coefficient ξ.

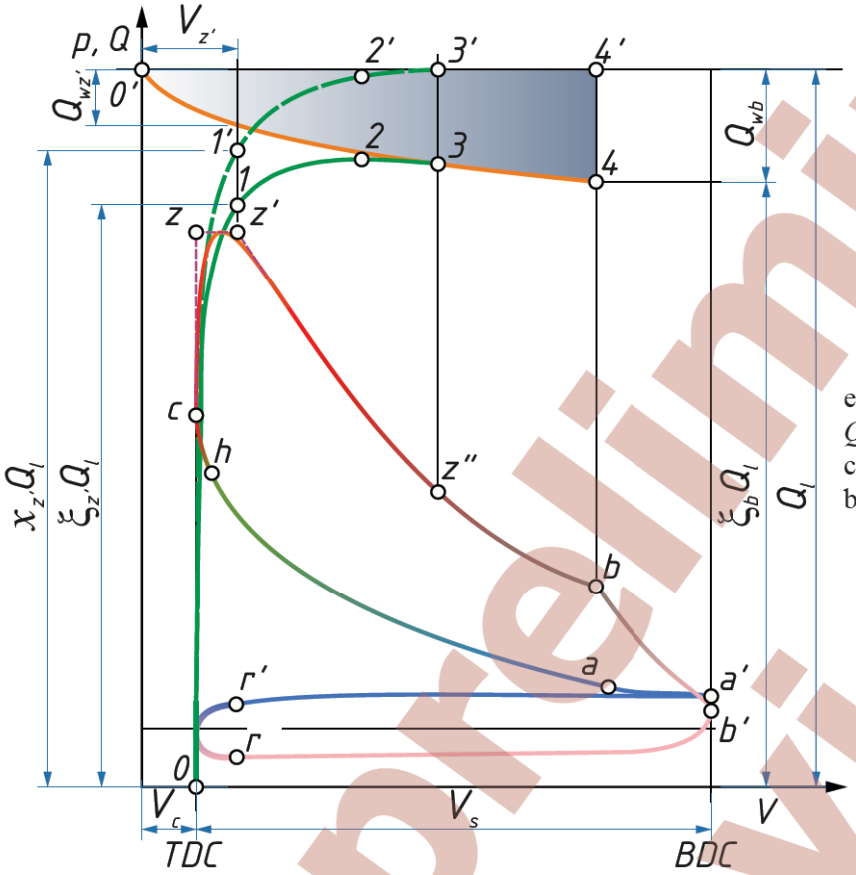


Figure 5.202 – Combustion and expansion processes in *pV*- and *QV* – coordinates in the operating cycle of a real engine with a combined heat supply

In calculations of operating processes, experimental values of the heat utilization coefficient corresponding to points *z* and *b* of the calculation cycle are used. For low- and medium-speed engines, the values of ξ_z lie in the range of 0.75...0.90, and ξ_b – 0.86...0.98. For high-speed engines, these values are respectively 0.70...0.85 and 0.85...0.95.

Combustion equation. If we assume that the combustion process occurs first at *V* = const, and then at *p* = const and ends at point *z′*, and neglect the insignificant difference between the heat capacities of the residual gases and combustion products, then the heat balance equation for the combustion process occurring in the section *czz′*, in accordance with the first law of thermodynamics, has the form:

$$x_{z'} Q_l - Q_{wz'} = U_{z'} - U_c + L_{zz'} ,$$

or in finite differences:

$$\Delta Q_{czz'} = \Delta U_{czz'} + \Delta L_{zz'} ; \qquad (5.13)$$

where $Q_{wz'}$ – heat loss into the cooled walls; U_c and $U_{z'}$ – internal energy of air and combustion products at points *c* and *z′*; $L_{zz'}$ – work of expansion of gases in the section *zz′*.

Let us accept the assumption that the thermal effect of the combustion reaction does not depend on the temperature of the reacting substances. Then the amount of heat released during 1 kg fuel combustion, which goes to increase the internal energy of the working fluid and perform external

For preliminary review

mechanical work in the section *czz′*, can be expressed as:

$$\Delta Q_{czz'} = \xi_{z'} Q_l \,. \tag{5.14}$$

The increment in the internal energy of the working fluid in the temperature range $T_{z'}...T_c$ can be written as the difference in internal energies at points *z′* and *c*:

$$\Delta U_{czz'} = U_{z'} - U_c = M_{z'} \overline{c}''_{v_{z'}} T_{z'} - M_c \overline{c}'_{v_c} T_c \,, \tag{5.15}$$

where $M_{z'}$, M_c – masses of combustion products, $\overline{c}'_{v_c}$ – average molar isochoric heat capacity of the mixture at point *c*; $\overline{c}''_{v_{z'}}$ – average molar isochoric heat capacity of the mixture at point *z′*.

The amount of combustion products $M_{z'}$ at point *z′* can be determined using the concept of *molecular change coefficient* β.

For any period of time from the start of the combustion process t, the calculated coefficient of molecular change can be determined taking into account the parts of fuel *x* burned up to this moment using the formula:

$$\beta_x = \frac{M_x + M_r}{L + M_r} = \frac{L + \Delta M_x + M_r}{L + M_r} = 1 + \frac{\beta_0 - 1}{1 + \gamma_r} x \,. \tag{5.16}$$

where M_x is the number of moles of gaseous products formed from the combustion of a parts of fuel *x*; *L* – number of moles of air excluding residual gases in the cylinder; M_r – the number of moles of residual gases remaining in the cylinder from the previous cycle; γ_r – coefficient of residual gases $\gamma_r = M_r / L$.

From expression (5.16) it follows that on the compression line and at the initial moment of ignition, i.e. at $x = 0$, $\beta_x = 1$, and at the end of the combustion process at $x = 1$ it reaches a maximum value equal to $\beta_{z'}$.

The amount of mixture at point *c* can be determined by the formula:

$$M_c = L(1 + \gamma_r) \,.$$

Then:

$$M_{z'} = \beta_{z'} M_c = \beta_{z'} L(1 + \gamma_r) \,,$$

Given these expressions:

$$\Delta U_{czz'} = \beta_{z'} L(1 + \gamma_r)\, \overline{c}''_{v_{z'}} T_{z'} - L(1 + \gamma_r)\, \overline{c}'_{v_c} T_c \,, \tag{5.17}$$

Average molar isochoric heat capacity of the working mixture at the end of compression (point *c*) with the fraction of burned fuel $x = 0$:

$$\overline{c}'_{\upsilon_c} = \frac{\gamma_r \overline{c}''_v + [\alpha(1 + \gamma_r - \gamma_r)]\, \overline{c}'_v}{\alpha(1 + \gamma_r)} \,, \text{ kJ/(kmol×K)}.$$

where $\overline{c}'_v$ is the average molar isochoric heat capacity of dry air (kJ/(kmol×K)) can be determined from the approximated dependence $\overline{c}'_v = 19{,}264 - 0{,}0025T$; $\overline{c}''_v$ – average molar isochoric heat capacity of residual gases (pure combustion products formed at α = 1). For fuel of average elemental chemical composition $\overline{c}''_v = 20{,}47 + 0{,}0036T$.

At the end of visible combustion (at point *z ′*), when $x = x_{z'}$ with $\alpha > 1$, a gas mixture consisting of pure combustion products and air not used during combustion is formed in the cylinder. The average molar heat capacity of such a mixture $\overline{c}''_{v_{z'}}$ with a fraction of burned fuel equal to x_z is determined according to the gas mixing rule as:

$$\overline{c}''_{v_{z'}} = \frac{(m x_{z'} + \gamma_r)\, \overline{c}''_v + [\alpha(1 + \gamma_r) - (x_{z'} + \gamma_r)]\, \overline{c}'_{v_c}}{\alpha\ (1 + \gamma_r) + (m - 1)\ x_{z'}} \,.$$

Pressure and temperatures p_c, $p_{z'}$, T_c and $T_{z'}$ are determined from the characteristic equations of

For preliminary review

state:

$$p_c V_c = 8{,}314 M_c T_c,$$
$$p_{z'} V_{z'} = 8{,}314 M_{z'} T_{z'},$$

where M_c is the number of moles of charge at the end of compression; $M_{z'}$ – number of moles of charge at the end of combustion.

Heat equivalent to the work of the isobaric process zz':

$$\Delta L_{zz'} = p_{z'} V_{z'} - p_z V_z = p_{z'} V_{z'} - \lambda p_c V_c = 8{,}314 M_{z'} T_{z'} - 8{,}314 \lambda M_c T_c = \\ = 8{,}314 \beta_{z'} L (1 + \gamma_r)\, T_{z'} - 8{,}314 \lambda L (1 + \gamma_r)\, T_c. \quad (5.18)$$

In these equations, the quantities $V_{z'}$ and $T_{z'}$ remain unknown, which are determined from the combustion equation.

Substituting expressions (5.14), (5.17) and (5.18) into equation (5.13) and performing transformations, taking into account that the relationship between isobaric and isochoric heat capacities $\overline{c}''_{p_{z'}} = \overline{c}''_{v_{z'}} + 8{,}314$, we obtain the combustion equation as:

$$\frac{\xi_{z'} Q_l}{\alpha L_0} + \left[\overline{c}'_v + 8{,}314\lambda + \gamma_r \left(\overline{c}''_v + 8{,}314\lambda\right)\right] T_c = \beta_{z'} (1 + \gamma_r)\, \overline{c}''_{p_{z'}} T_{z'}. \quad (5.19)$$

where L_0 is the theoretically required amount of dry atmospheric air to burn one kilogram of fuel, kmol/kg.

Equation (5.19) is usually solved by the method of successive approximations. As a first approximation, they are specified by the temperature value within the range $T_{z'} = 1700\ldots2000$ K.

As a result of substituting into equation (5.19) all the numerical values of the known quantities $\xi_{z'}$, Q_l, γ, L_0, $\overline{c}'_v$, $\overline{c}''_v$ and $\lambda = p_z / p_c$, as well as the dependence $\overline{c}''_{p_{z'}}$ on the desired temperature $T_{z'}$, the combustion equation is reduced to a quadratic equation type:

$$A T_{z'}^2 + B T_{z'} - C = 0;$$

$$T_{z'} = \frac{-B + \sqrt{B^2 + 4AC}}{2A}, \text{ K}.$$

The temperature $T_{z'}$ at which the parts of the equation converge is the desired one.

For marine diesel engines, the temperature values at the end of visible combustion $T_{z'}$ lie within the following limits: low-speed 1600…1800 K; medium speed 1700…1900 K; high-speed 1800… 2000 K.

The volume of the cylinder at the end of visible combustion is determined depending on the degree of pre-expansion: $V_{z'} = \rho V_s$.

The degree of preliminary expansion can be determined as a result of the joint solution of the equations of state of the gas at points z' and c. Dividing the characteristic equations of state term by term and transforming the resulting equality taking into account the relations $\lambda = p_{z'} / p_c$; $\rho = V_{z'} / V_c$; $\beta_z = M_{z'} / M_c$, we obtain a formula for calculating the degree of preliminary expansion: $\rho = \beta_{z'} T_{z'} / (\lambda T_c)$. At nominal mode for most diesel engines $\rho = 1.2\ldots1.6$. The degree of preliminary expansion depends on the method and quality of the mixture formation processes, fuel combustion, fuel injection advance angle, load and diesel engine speed.

Indicators of the dynamism of the operating process. To assess the nature of changes in loads on engine structural elements during fuel combustion, indicators of the dynamism (rigidity) of the operating process are usually used: the average and maximum rate of pressure increase.

The increase in pressure Δp in the cylinder at the angle $\Delta\varphi$ which corresponds to the period of initial combustion (θ_{II}) is called the average rate of pressure increase during combustion, $\omega_{av} = \Delta p / \Delta\varphi$, MPa/deg. (Fig. 5.203). The moment of combustion onset can be determined with a sufficient degree of accuracy as the inflection point of the function $dp/d\varphi$ (point 1 in Fig. 5.201) af-

For preliminary review

ter which it begins to increase.

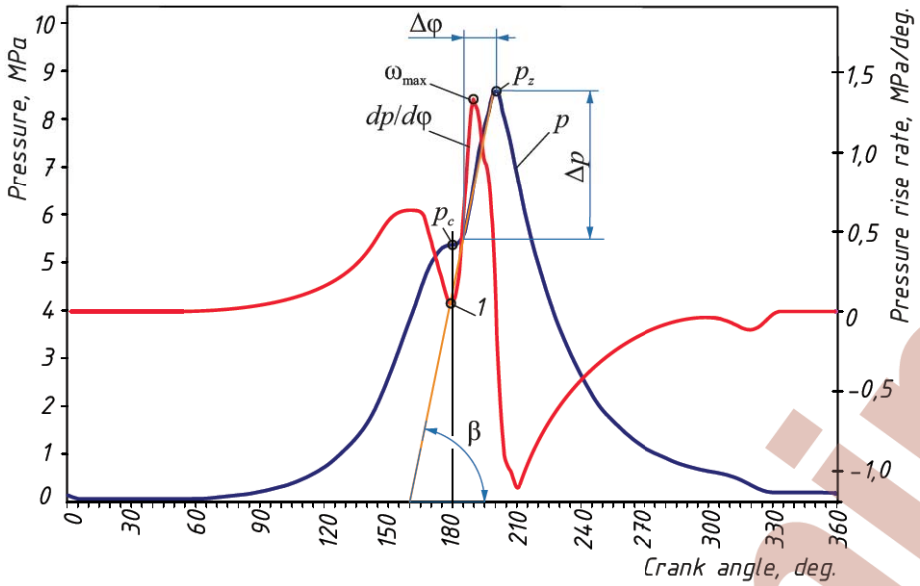


Figure 5.203 – Expanded indicator diagram in $p\varphi$- coordinates and a diagram of the rate of pressure change ($dp/d\varphi$) constructed from it (adapted from [116])

The maximum rate of pressure increase during combustion is determined from the curve of the rate of pressure change $dp/d\varphi$ (the first derivative of the pressure change with respect to the angle of rotation of the crank, Fig. 5.203) at the steepest section of the pressure increase in the cylinder $\omega_{max}=(\Delta p/\Delta\varphi)_{max}$. In practice, the value of ω_{max} can be determined from the indicator diagram as the tangent of the tangent angle to the steepest section of the pressure curve, as shown in Fig. 5.203:

$$\omega_{max}=(m_p/m_\varphi)\tan\beta$$

Where m_p – pressure scale, MPa/mm; m_φ – scale of rotation angles, deg/mm; β – angle of inclination of the tangent, degrees.

To a large extent, the rate of pressure increase (the dynamism of the operating process) depends on *the dynamism factor,* which is the ratio of the mass of fuel injected during the auto-ignition delay period τ to the full cyclic portion of fuel $\sigma = G_{\tau i}/G_c$ (Fig. 5.199).

The smaller the portion of fuel $G_{\tau i}$ supplied during the period φ_τ, the lower the mechanical stress of the engine.

As the dynamism factor increases, the dynamism of the operating process increases accordingly. For low- and medium-speed diesel engines at rated power modes, the values of σ are in the range of 0.4...1.0, and ω_{av} usually lies in the range of 0.2... 0.6 MPa/deg.

For high-speed forced diesel engines, the dynamic factor is σ = 0.8...1.0, and ω_{av} can reach 0.80...1.4 MPa/deg, which significantly increases the load on the parts of the piston group and the crank mechanism. An increase in the dynamics of the operating process is usually observed when the conditions for self-ignition of the fuel worsen.

The rate of pressure increase during fuel combustion is reduced by reducing the value of the dynamic factor. In practice, this is achieved either by changing the injection characteristics or by reducing the auto-ignition delay period.

Injection characteristics have a significant impact on the dynamics of the operating process. For modern electronically controlled fuel systems, the characteristic depends on the implemented algorithm for supplying fuel to the combustion chamber.

With an increase in the dynamic factor (for example, by appropriate profiling of the cam washer of the injection pump drive), a large mass of fuel is injected into the combustion chamber during the self-ignition delay period φ_τ. As a result, the rigidity of the operating process increases, the maximum amount of heat is released in the TDC region, the maximum cycle pressure increases, and the efficiency of the operating process increases.

As the dynamic factor decreases, the maximum cycle pressure p_z and the rate of pressure increase during combustion ω_{av} decrease, and the duration of fuel supply increases. As a result, the

For preliminary review

angle φ_z from TDC increases until p_z is reached, the period of fuel burning becomes longer, and engine efficiency deteriorates.

The process of its combustion depends on the characteristics of fuel injection in the final phase. When designing fuel equipment, they strive to reduce the proportion of fuel in this phase by sharply cutting off the supply.

Recently widespread accumulator fuel injection systemsand systemswith hydraulically driven fuel pumps controlled by microprocessor technology allow the implementation of more complex fuel supply algorithms compared to mechanical injection systems. For example, the method of dividing the cyclic supply into several different-sized portions supplied to the combustion chamber in a certain sequence has become widespread. This makes it possible to effectively influence the dynamism of operating processes to achieve the specified indicators of efficiency, environmental friendliness and engine reliability. We will consider new fuel supply control algorithms in the following sections.

Dynamics of heat release during combustion. Important indicators of the fuel combustion process are the dynamics of heat release in the engine working space. There are integral and differential characteristics of heat release (Fig. 5.204).

The total amount of heat that is released at the current moment of the combustion process (Q_i) is called the integral characteristic of heat release. Defined as a function of time or rotation angle.

The amount of heat that is released at the current value of the crankshaft rotation angle ($dQ_i/d\varphi$ or dQ_i/dt) is called the differential heat release characteristic. In fact, it is the first derivative of the integral characteristic and characterizes the rate of heat release in the engine working space.

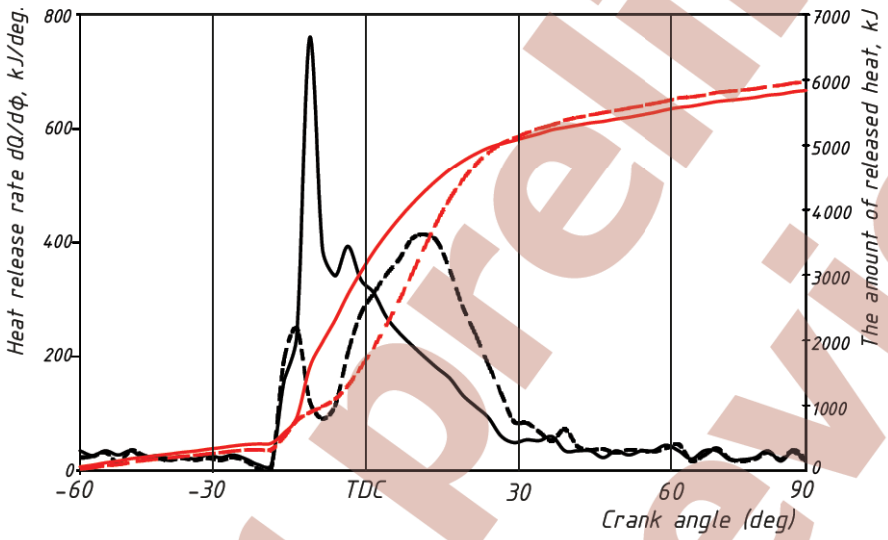


Figure 5.204 – Heat release rate (black lines) and the amount of heat released (red lines) as a function of the engine crankshaft rotation angle when fuel is supplied in one portion (solid lines) and with pilot injection (dashed lines) (adapted from [9])

The amount of heat released during fuel combustion and the rate of heat release in the diesel cylinder are determined based on the analysis and processing of indicator diagrams.

To compare combustion processes in different engines that differ in size and design, dimensionless indicators of heat release dynamics are often used. In this case, all the heat released during the combustion process is taken as one, and the integral characteristic shows the fraction of heat x released up to the current moment. The differential characteristic characterizes the relative rate of heat release, $dx/d\varphi$.

The nature of heat release during the combustion process depends on structural, physical and chemical factors: the design of the combustion chamber, the advance angle of fuel injection, the fuel injection algorithm into the combustion chamber, engine operating mode, the quality of fuel atomization and mixture formation, pressure and temperature of the charge at the end of compression, concentration oxygen in the cylinder during combustion, the ability of the fuel to self-ignite.

The main characteristics of fuel supply and combustion, as well as the associated dependencies of changes in pressure and temperature in the working cylinder of a diesel engine are presented in Fig. 5.205.

For preliminary review

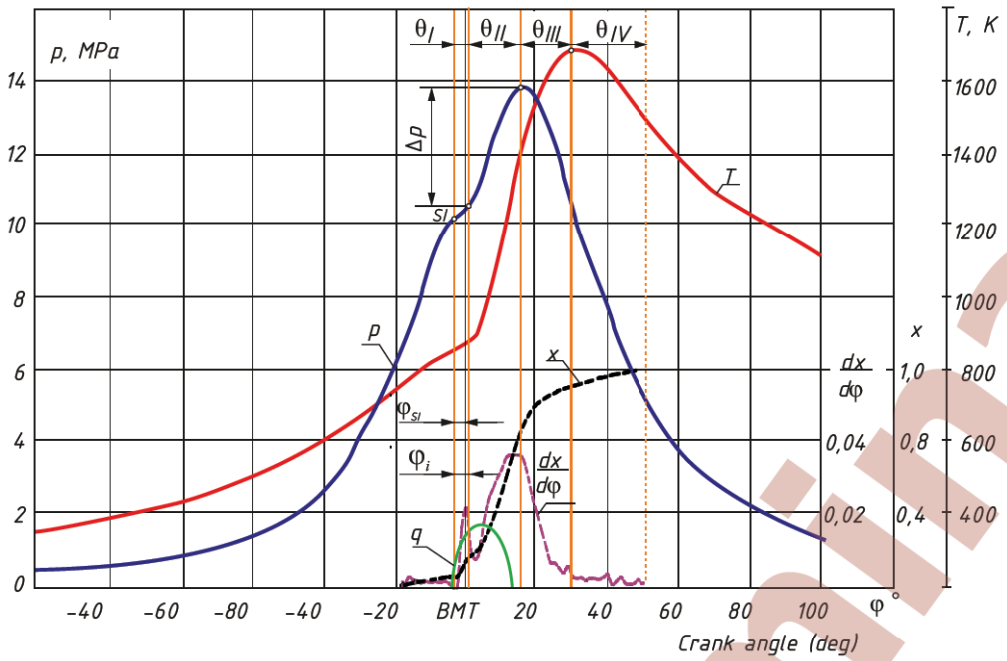


Figure 5.205 – Diagram of changes in operating process parameters in a diesel cylinder

As has already been shown above, the main parameters of fuel supply, which have the greatest significant impact on the operating process of a diesel engine, are: the quality of fuel atomization, injection characteristics, method of mixture formation, etc. However, for organizing the operating process, not only quantitative and qualitative indicators of the fuel supply process are extremely important, but also the connection of the fuel supply process to the position of the piston in the working cylinder of the engine on the compression stroke.

5.9.9 The influence of fuel supply characteristics on the diesel operating process

As already noted, the initial stage of fuel entry into the combustion chamber before its ignition and combustion, which corresponds to the *delay period* of fuel self-ignition φ_τ, has a significant influence on the process of fuel combustion in the working cylinder.

On the expanded indicator diagram of the diesel operating process (Fig. 5.206), the duration of this period is defined as the angular interval from the moment the first portions of fuel enter the working cylinder (point 1) until the moment the combustion line separates from the compression line (point 2). The compression line means the pressure change curve in the working cylinder in the absence of fuel supply, the combustion line means the pressure change curve during fuel combustion.

During the self-ignition delay period, a series of sequential-parallel physico-chemical processes, called *pre-flame processes, occur*.

When the first portions of fuel enter the working cylinder, part of the heat of the charge is spent on heating and evaporating them. As a result, the temperature and pressure in the cylinder decrease slightly, the combustion curve goes below the compression curve (position A in Fig. 5.206).

As the fuel evaporates, chemical reactions begin to form primary complexes, called *pre-flame reactions*. These reactions can be either exo- or endothermic in nature. Only after the accumulation of primary reaction products in the combustion chamber does their interaction with atmospheric oxygen begin, which, as a rule, is of a chain nature and is accompanied by the release of a large amount of heat. An increase in charge temperature leads to an increase in pressure, as a result of which the combustion curve intersects the compression curve, which corresponds to the end of the auto-ignition delay period.

The duration of the self-ignition delay period is mainly determined by the temperature of the charge at the time of fuel injection, the properties of the fuel itself, and the quality of its atomization. The latter largely depends on the performance of the fuel equipment.

To obtain a given pattern of pressure changes in the working cylinder, it is necessary to take into

For preliminary review

account the time required for pre-flame processes. To do this, the moment of the start of fuel supply is set earlier than the theoretically determined moment of the start of heat release by the auto-ignition delay angle (φ_{fa}). In practice, the influence of the self-ignition delay period on the operating process is taken into account by setting the static advance angle of the fuel pump supply φ_{pdt} (Fig. 5.196).

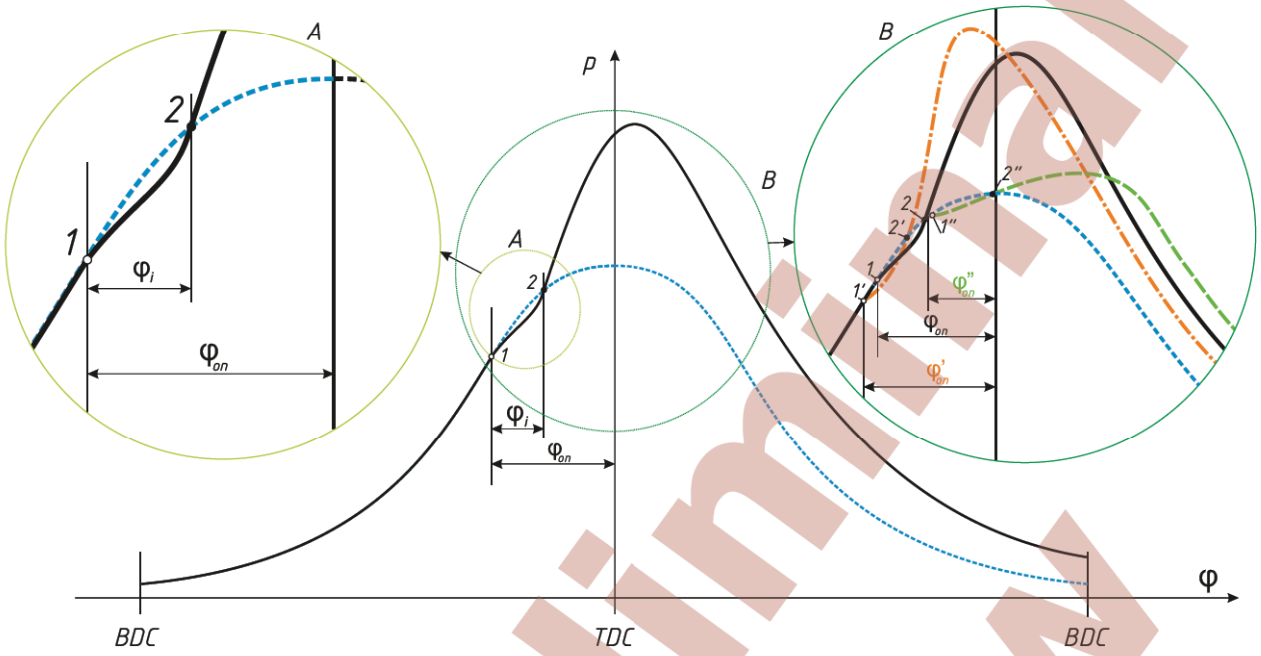


Figure 5.206 – Self-ignition delay period and its effect on the engine's operating process: 1 – the moment the supply starts; 2 – moment of separation of the combustion line from the compression line. – – – – – compression line; —— – combustion line at the optimal supply advance angle; – · – · – combustion line during early fuel injection; – – – – combustion line with late fuel injection (adapted from [9])

With an increase in φ_{pdt}, fuel is injected into the cylinder earlier (point 1′ in Fig. 5.206), which leads to its earlier ignition. As a result, a greater amount of heat is released even before the piston reaches TDC, which leads to a sharper increase in pressure and an increase in its maximum value. The operating process becomes more dynamic and more rigid. With a further increase in the advance angle, this tendency will weaken, since the fuel will be injected into an environment with a lower temperature and pressure, and this will lead to an increase in the auto-ignition delay period.

With an increase in φ_{pdt}, the efficiency of a diesel engine initially increases, since a certain increase in the compression work up to TDC is more than compensated by an increase in the thermal efficiency of the cycle due to the supply of heat to the working fluid at a higher temperature. At large values of the angle φ_{pdt}, the compression work increases significantly and becomes greater than the gain in thermal efficiency, so the diesel efficiency decreases.

With a decrease in the angle φ_{pdt}, especially to the values corresponding to the beginning of fuel combustion after TDC (point 1″ in Fig. 5.206), the mechanical stress of the engine decreases, but at the same time its efficiency decreases. The combustion of the main portion of the fuel shifts to the expansion line, which increases the temperature of the exhaust gases and the thermal stress of the parts of the cylinder-piston group.

Obviously, the injection advance angle must increase with increasing engine speed in order to provide the necessary time period for pre-flame processes to occur. In addition, changes in engine load, boost pressure, external conditions, and fuel grade may require adjustments to the supply advance angle.

The advance angle is an important parameter affecting the performance of the operating process, engine efficiency, and its environmental performance. In this regard, the bulk of the fuel systems of modern marine diesel engines are equipped with devices for automatically changing this parameter depending on the engine operating mode. We discussed the design of such systems in previous sec-

For preliminary review

tions. It should be noted that the principle of choosing the optimal advance angle can be most fully realized only in systemswith electronic fuel supply control.

A number of modern high- and medium-speed diesel engines provide for a change in the nature of the operating process depending on the load-speed regime. In particular, the transition from the classical cycle with mixed heat supply at low and medium load modes to the Miller cycle at load modes close to maximum.

This transition is accompanied by a simultaneous change in valve timing and fuel supply. In Fig. 5.207 presents a variant of a technical solution that allows such a transition, which was developed by MaK and implemented in engines of the M20-M43 series.

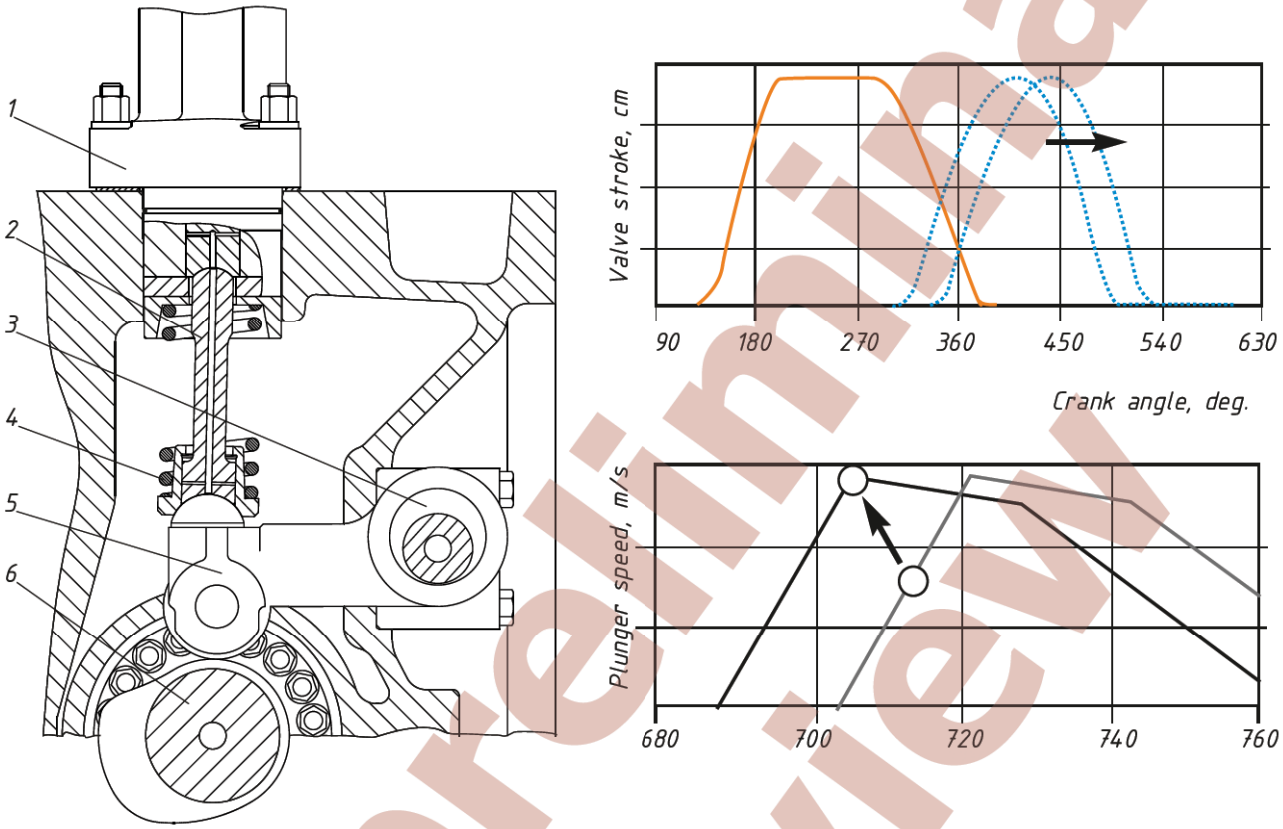


Figure 5.207 – Mechanism for changing the injection advance angle and valve timing of MaK engines: 1 – injection pump; 2 – push rod; 3 – eccentric; 4 – return spring; 5 – lever pusher; 6 – camshaft (adapted from [24, 117])

The operating principle of the device is based on changing the position of the lever pusher roller relative to the camshaft cam washer. To do this, the lever axis is mounted eccentrically on a shaft, which has the ability to rotate through an angle close to 180°. As a result, the pusher makes a translational movement, changing the advance angle of the fuel supply and the angles at which the intake valve begins to open and close.

The eccentric shafts of the roller pushers are driven by a pneumatic servomotor through a gear system. Manual switching of the engine from one mode to another is also provided.

Changing the advance angle in this design allows not only to ensure the optimal heat release law at maximum power mode, but also to improve the conditions for fuel atomization when the load is reduced by shifting the start of injection to a higher-speed section of the plunger lift.

In Fig. 5.208 shows a device for changing the supply advance angle used by MAN in its medium-speed engines. In this device, the pump drive shaft is connected to the drive gear through an inclined spline connection. During axial movement, the gear hub slides along the splines and rotates the camshaft relative to the crankshaft at a certain angle, the magnitude of which is determined by the angle of inclination of the splines to the shaft axis and the amount of axial movement. For axial movement of the gear together with the hub, a hydraulic servo drive is used, located at the end of the shaft on the engine crankcase.

For preliminary review

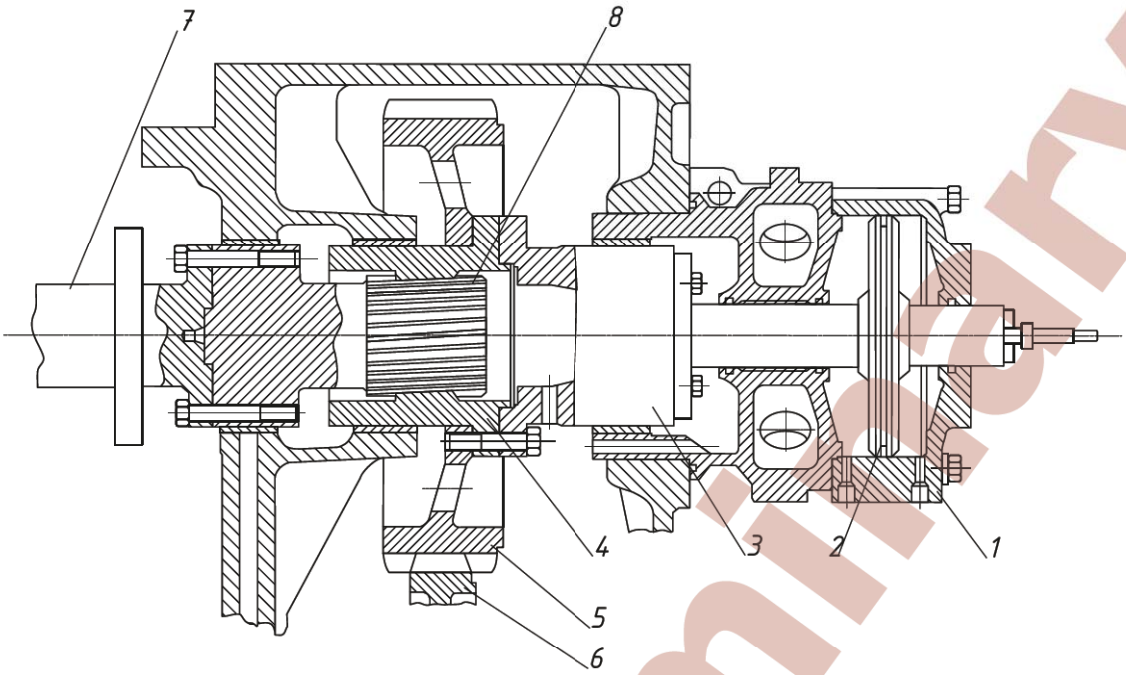


Figure 5.208 – Mechanism for changing the injection advance angle of medium-speed MAN engines: 1 – servo cylinder; 2 – servo piston; 3 – drive gear shaft; 4 – hub; 5 – drive gear; 6 – intermediate gear; 7 – pump drive shaft; 8 – sliding spline connection with inclined splines (adapted from [118])

On high-speed engines that operate at different speed modes, automatic centrifugal-type injection advance clutches are used. They are designed to automatically change the fuel injection advance angle when the engine crankshaft speed changes. The operation of such a coupling is shown schematically in Fig. 5.209. In the coupling housing, through which it is driven, a half-coupling is mounted, through which the block-type fuel injection pump shaft is driven. The coupling half has protrusions, which on one side are loaded with springs, and on the other side rest against eccentrics made on unbalanced weights. Thus, the coupling half occupies a certain position relative to the housing. As the rotation speed increases, centrifugal force begins to act on the unbalanced part of the weights. Under the influence of this force, the weights, overcoming the force of the springs, move apart and through the eccentrics, turning the coupling half at an angle γ against the direction of rotation of the housing, thereby changing the supply advance angle.

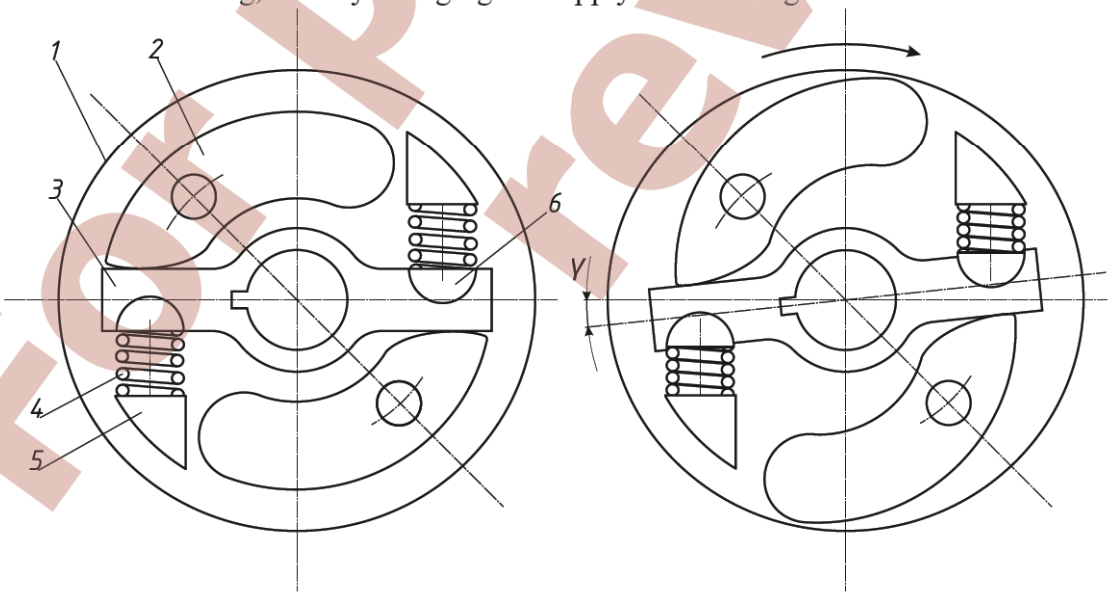


Figure 5.209 – Centrifugal clutch for changing the injection advance angle of a high-speed diesel engine depending on its rotation speed: 1 – coupling housing; 2 – unbalanced weights; 3 – half coupling; 4 – spring; 5 – fixed stop; 6 – movable stop (adapted from [9])

For preliminary review

The injection duration (angle φ_{edt}) also has a great influence on the operating process. To increase efficiency and reduce exhaust gas temperature, it is necessary to ensure a relatively small value of the angle φ_{edt} at nominal mode. This angle can be reduced by increasing the maximum injection pressure or increasing the effective fuel injector flow area. In the first case, the mechanical loads on the parts of the fuel equipment will increase, and in the second, at low load conditions there will be a low injection pressure, which will lead to worse fuel atomization.

5.9.10 Current trends in the organization of engine operating processe

The introduction of new types of fuel systems and, first of all, electronic fuel supply control systems on marine diesel engines has made it possible to optimize operating processes in relation to given operating conditions. In this regard, it seems interesting to highlight some ways to improve the operating process of engines by optimally choosing the law of fuel supply to the cylinder.

The trend of increasing injection pressures that has emerged in recent years is aimed at reducing the supply period and increasing the completeness of fuel combustion.

In Fig. 5.210 *a* shows the injection process with a sharp increase in pressure in the initial stage and a subsequent decline, accompanied by a decrease in the rate of fuel entry into the cylinder. This implementation of the injection law is typical for highly accelerated direct-acting fuel systems and accumulator systems with small accumulator volumes. High pressure in the initial stage contributes to the high rate of expiration of the first portions of fuel entering the combustion chamber. As a result, the first portions are well atomized, distributed throughout the entire volume of the combustion chamber, pre-flame processes proceed faster, and the self-ignition delay time is reduced.

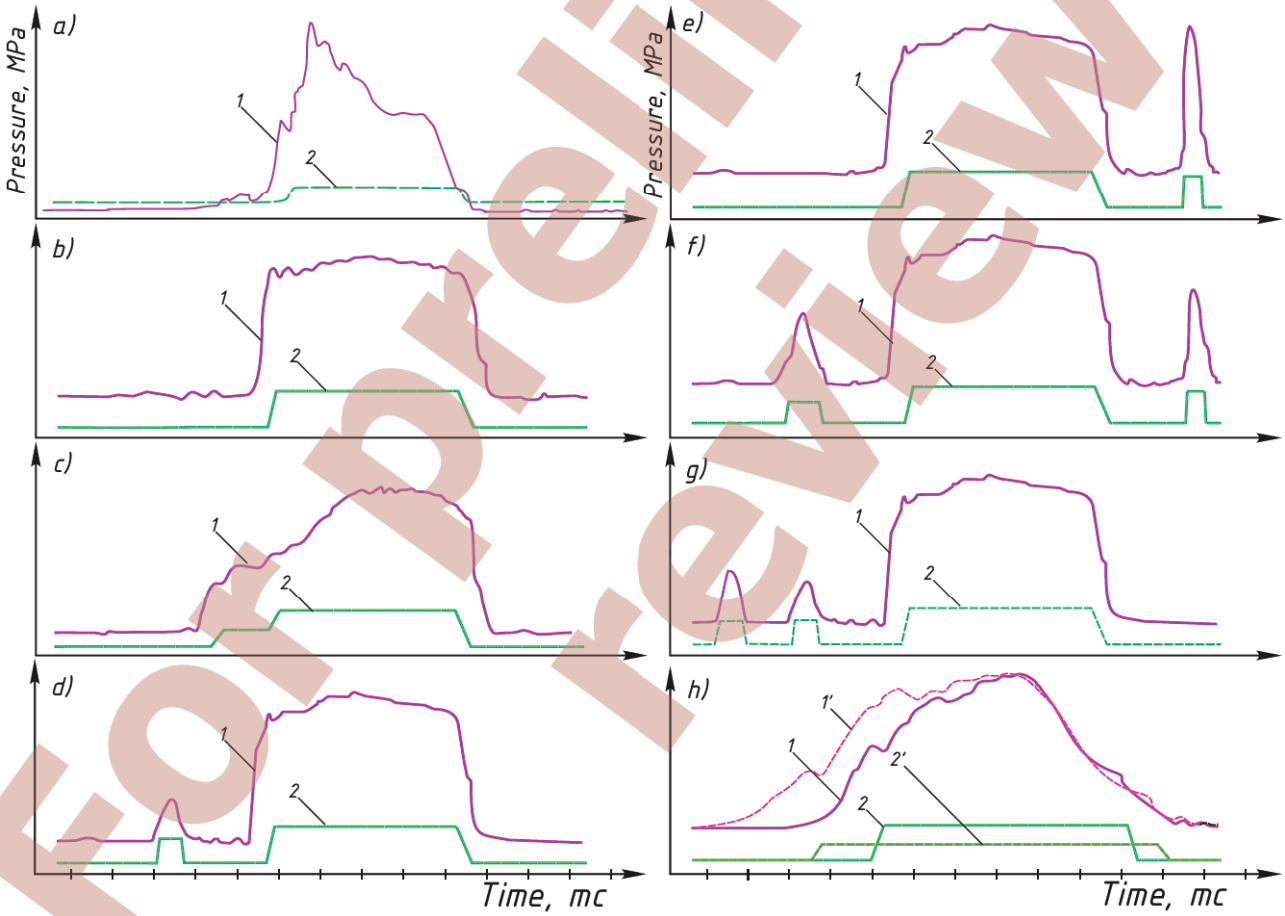


Figure 5.210 – Variants of injection laws implemented in modern diesel engines: *a* – with a rapid increase in pressure in the initial stage; *b* – with constant pressure; *c* – with a smooth increase and sharp release of pressure; *d* – two-phase injection with pre-injection; *e* – two-phase injection with post-injection; *f* – three-phase injection with pre-injection and post-injection; *g* – three-phase injection with two pre-injections; *h* – parallel injection of pilot and main portions of fuel; 1 – pressure in front of the fuel injector; 2 – lifting the spray needle (adapted from [9, 36, 44])

For preliminary review

The ignition and combustion of the first portions leads to the fact that subsequent fuel, despite the poorer atomization quality, enters an environment with elevated temperature and pressure, in which there are already pockets of open flame. This promotes rapid evaporation of subsequent portions of fuel and intense combustion. This organization of the operating process led to an increase in engine efficiency. However, the concentration of the entire combustion process in a narrow section of the cycle has led to an increase in combustion severity, an increase in the noise level generated by the engine, and an increase in loads on the parts of the cylinder-piston group and the crankshaft.

The second version of the supply law with constant injection pressure is inherent in accumulator systems with a large accumulator volume, when the loss of a portion of fuel equal to the cyclic supply does not lead to a significant drop in pressure in the system (Fig. 5.210 *b*). This method is characterized by the fact that throughout the entire injection the quality of atomization remains consistently high, as a result of which pre-flame processes and subsequent combustion proceed quite quickly. As in the first case, with such an organization of the injection process, high engine efficiency is achieved, but large thermal and mechanical loads arise on the CPG and crankshaft.

To avoid the noted phenomena in a number of modes, when the time factor allotted for fuel combustion is not so critical (for example, in LSE), a gentle law of increase in injection pressure is used in the initial stage (Fig. 5.210 *c*). This supply law is implemented by transferring the active stroke of the plunger to a section with lower acceleration, for example, by changing the valve opening angles of valve-type pumps or by changing the position of the pusher roller relative to the fuel injection pump drive cam. This law is most simply implemented in engines with hydraulically driven fuel pumps or in accumulator injection systems with electronic fuel control. For example, in Mitsubishi UEC-Eco series engines, the use of such a law is considered as a compromise solution between the efficient and environmental performance of the engine when operating at low speeds, but with a high load. The influence of the supply law with a gentle increase in pressure on the operating process of a low-speed engine of the ME series from MAN is shown in Fig. 5.210 *a*.

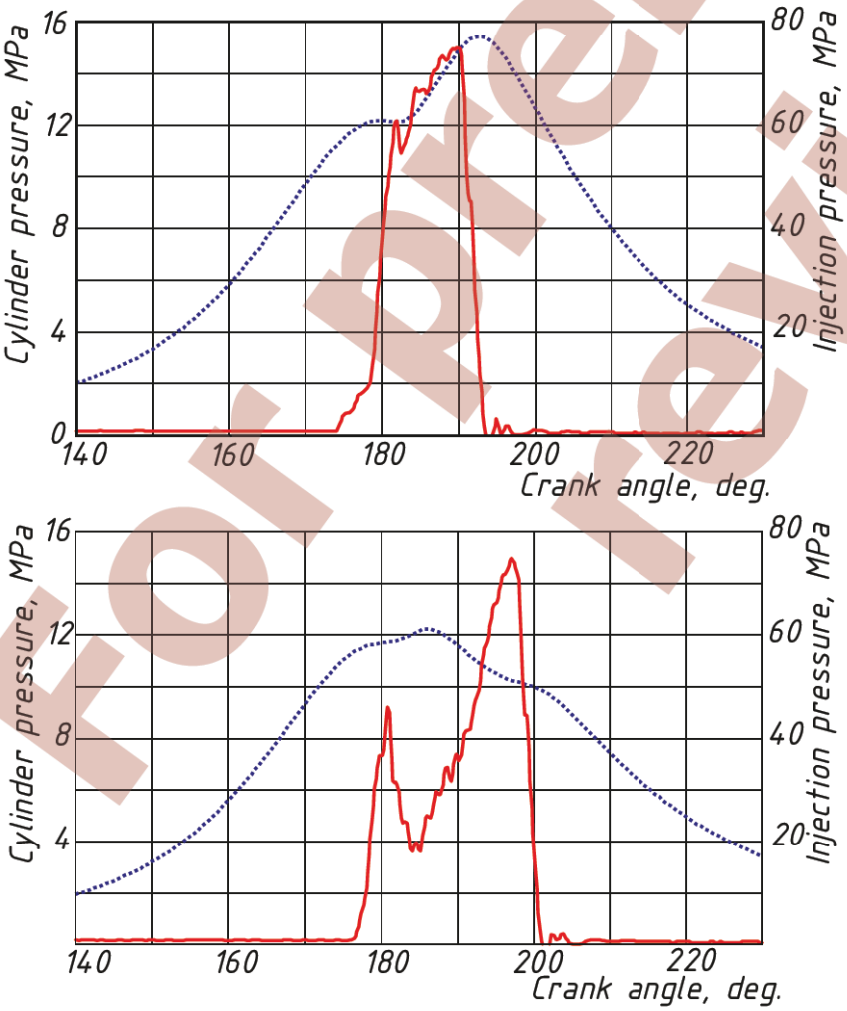


Figure 5.211 – Operating process of a low-speed engine with a gentle increase in injection pressure in the initial stage (*a*) and with two-phase fuel injection (*b*): —— fuel pressure in front of the fuel injector; - - - – cylinder pressure (adapted from [115])

For preliminary review

With a gentle increase in pressure in the initial stage, the dynamic factor of the injection process decreases. By the time of self-ignition of the fuel, a small part of the fuel from the full cycle supply is in the working cylinder. Its ignition and combustion are not able to lead to a sharp increase in pressure and temperature, but subsequent portions of fuel are injected into an environment with elevated temperature and pressure, which contributes to their rapid evaporation and combustion. The fuel burns as it enters the cylinder, that is, in fact, the duration of the combustion process is determined by the duration of the injection process. Such combustion proceeds smoothly, without sudden surges in pressure and temperature. As a result of extended heat generation, engine efficiency is slightly reduced, and environmental performance is improved. It is especially important that with such an organization of the operating process, local temperatures in the combustion chamber are reduced, since it is in areas with high local temperatures that the formation of nitrogen oxides, the most toxic component of diesel exhaust gases, occurs. At the same time, fuel atomization at the end of the supply is improved, and, consequently, more complete afterburning is achieved in the expansion line, which leads to a decrease in the content of soot particles in the exhaust gases. The flat injection law is also characterized by a reduction in the rigidity and noise of the diesel engine, and therefore a reduction in the load on its parts.

A lower rate of pressure rise in the initial stage and lower operating cycle temperatures are achieved with two-phase injection (Fig. 5.210 *d*, 5.211 *b*). This option is resorted to when the task is to reduce noise and reduce the NO_x content in exhaust gases.

This supply law is implemented by organizing pre-injection, when a small portion of fuel is injected into the cylinder before the main supply begins. To avoid deterioration in efficiency, the pilot portion of fuel in such a process should be as small as possible. Therefore, an important requirement for fuel equipment is its ability to provide a minimally stable supply of a pilot portion of fuel and the time interval between it and the main supply.

The presence of pre-injection creates conditions for a softer and more complete combustion, since the main supply occurs at the moment when a flame appears in the cylinder from the burning pre-fed portion of fuel. Therefore, ignition of the main mass occurs almost instantly and the combustion process proceeds at significantly lower speeds, determined by the rate of fuel entering the cylinder.

The nature of the operating process with two-phase fuel injection in a MAN ME series engine is shown in Fig. 5.211 *b*.

In systemswith electronic control of the fuel supply process, the absence of a rigid algorithm for controlling the injection law makes it possible to change the nature of the working cycle during engine operation, depending on the task at hand. If the goal is to achieve maximum engine efficiency, the injection law shown in Fig. 5.211 *a*. If it is necessary to reduce the content of harmful emissions as much as possible, the injection law shown in Fig. 5.211 *b*. The transition from one mode to another is carried out by changing the control algorithm, which requires a time within 0.25 s, that is, commensurate with the interval between individual cycles. In Fig. 5.212 shows the nature of the change in NO_x emissions during the transition of an ME series engine from the maximum efficiency mode to the maximum environmental friendliness mode.

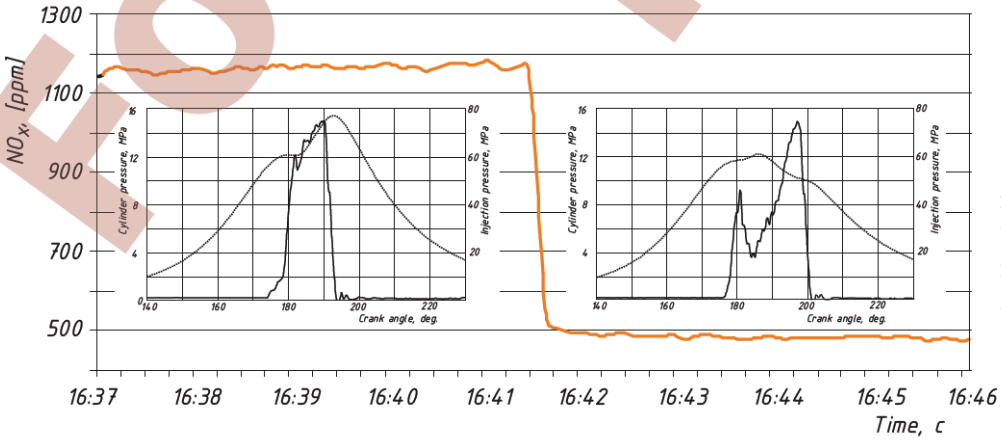


Figure 5.212 – Transition of an ME series engine from an economical operation mode to a mode that ensures a minimum content of harmful emissions in exhaust gases (adapted from [115])

For preliminary review

In this case, the maximum cycle pressure decreases from 15.7 to 12.2 MPa. At the same time, there is a reduction in NO_x emissions when switching to pre-injection mode from 1200 to 480 min^{-1}.

In Fig. 5.213 shows the dependences of specific fuel consumption, compression pressure and maximum pressure in the cylinder, as well as the content of nitrogen oxides in the exhaust gases as a function of engine load.

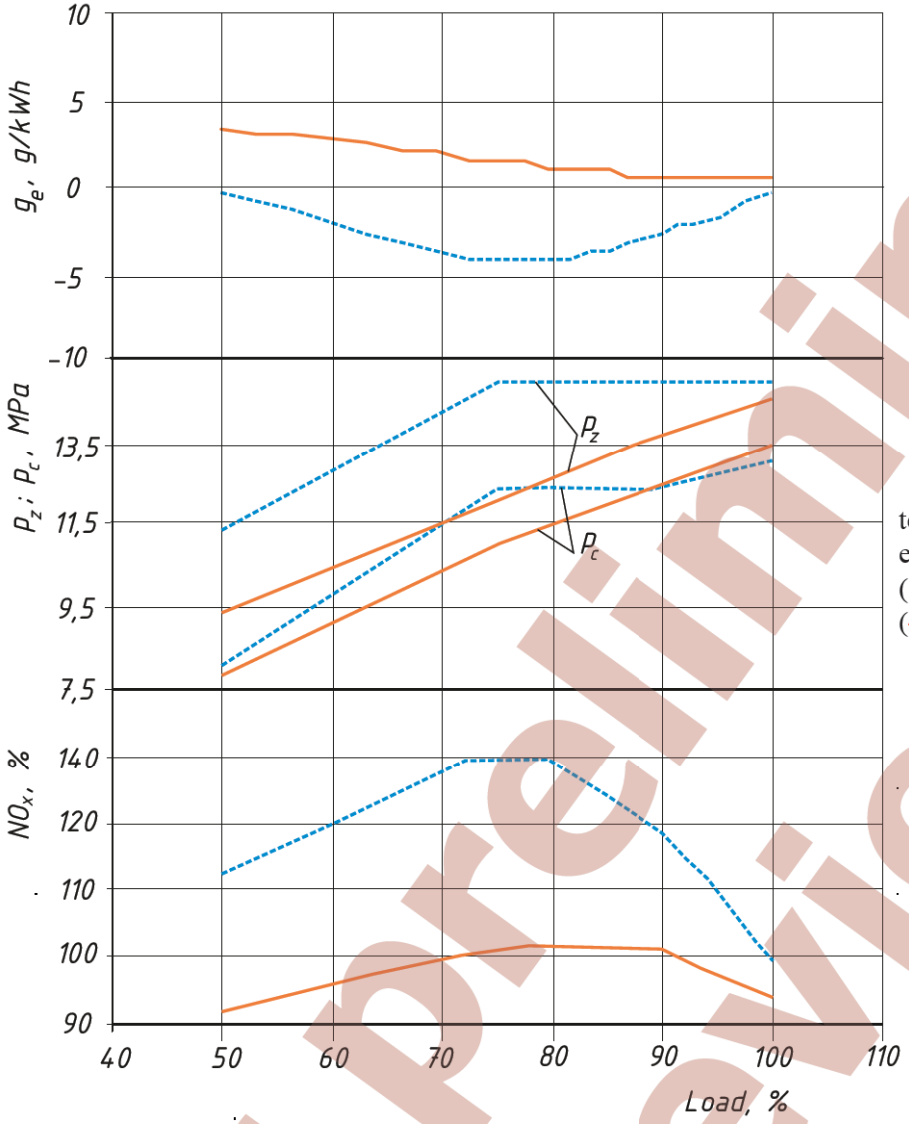


Figure 5.213 – Main characteristics of the MAN 7S60ME-C engine with a flat fuel supply (- - - -) and two-phase injection (———) (adapted from [115])

Due to the significant deterioration in the quality of marine diesel fuel over the past two decades, Sulzer has made a number of design changes to the fuel system of its RTA series engines. In addition to the VIT system already discussed above, the engines are equipped with a system for adjusting the fuel injection advance angle depending on its cetane number. This system is called Fuel Quality Setting (FQS).

The first generation of these systemsused the principle of manually adjusting the fuel supply depending on the quality of the fuel taken on board. Based on the engine indication results, the mechanic was able to set the required initial advance angle, from which the VIT system regulated the change in advance depending on the engine load (Fig. 5.214).

From the presented graph it can be seen that the FQS system allows you to change the initial angle of advance of the start of supply by 0.0...1.02 degrees throughout the entire range of operation of the VIT system.

In later developments, manual control was replaced by automatic control, which was carried out from an electronic control unit that received information from operating process control sensors. Retaining the possibility of manual control, the company retrofitted the engine control station with a pneumatic servo drive connected to an electronic unit.

For preliminary review

The combined use of VIT and FQS systemscan significantly expand the possibilities of influencing the operating process.

In operation, the joint use of the VIT system and the FQS system makes it possible to solve two main problems– to ensure maximum diesel efficiency or minimum emissions of harmful substances (primarily nitrogen oxides) in the exhaust gases (Fig. 5.215).

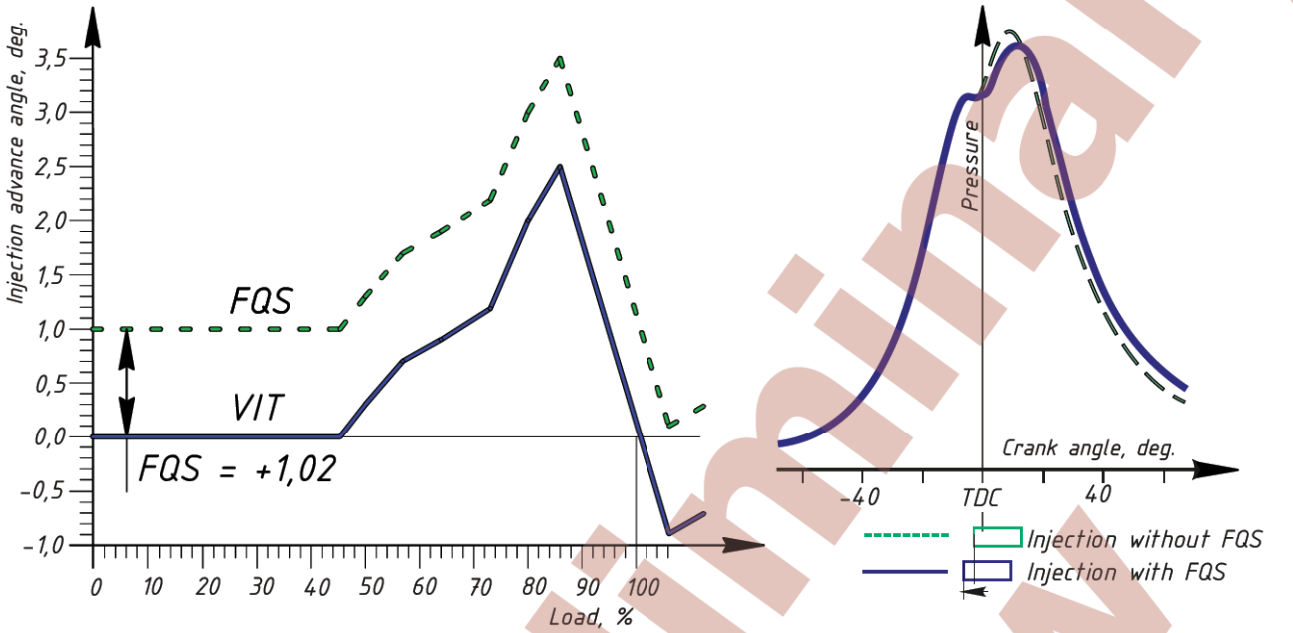


Figure 5.214 – Graph of joint operation of VIT and FQS systemsof RTA series engines. B influence of the FQS system on the diesel operating process (adapted from [32])

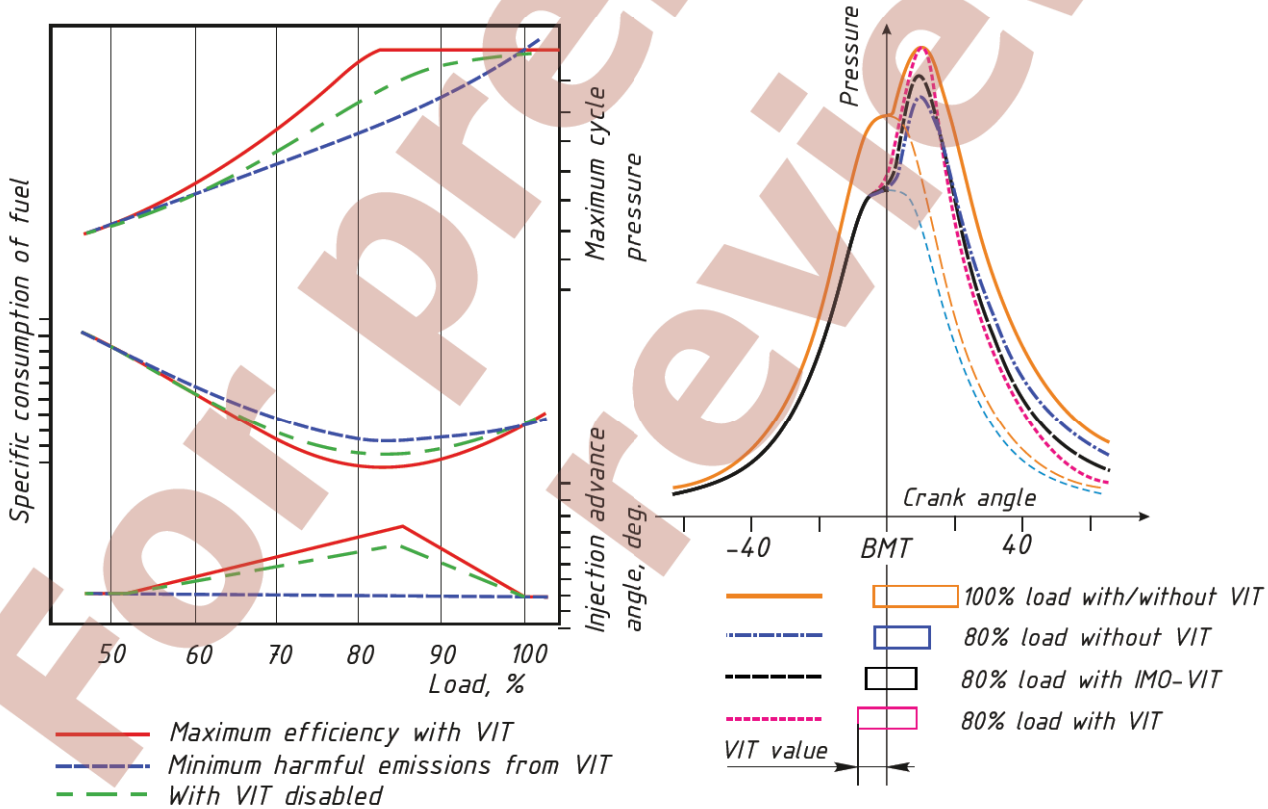


Figure 5.215 – The influence of the VIT and FQS systemson the characteristics and operating process of RTA series engines: *a* – the nature of the change in the maximum cycle pressure, specific fuel consumption and injection timing as a function of the load, depending on the method of organizing the operating process; *b* – influence of the VIT system on the diesel engine operating process (adapted from [32])

For preliminary review

To reduce exhaust smoke, some manufacturers resort to supplying a small additional portion of fuel at the end of combustion, when the main supply has already completed (Fig. 5.210 *e*). The combustion of a fresh portion of fuel intensifies the afterburning processes of the remaining free carbon particles and thereby reduces their content in the exhaust gases. A three-phase fuel supply process with pre-injection and post-injection is also considered promising (Fig. 5.210 *f*). Some engines use not one, but several pre-injections, which makes it possible to find a reasonable compromise between the efficient and environmental performance of a diesel engine (Fig. 5.210 *g*).

Multiphase injection is most simply implemented in accumulator injection systemsor in systemswith hydraulically driven plungers. Most often, the pilot portion is formed by submitting a short control signal to an electrically controlled fuel injector, however, cases of spool control of two-phase supply are also known. Thus, in its high-speed diesel engines of the 3126 series, Caterpillar used a plunger design with a spool edge to interrupt the flow (Fig. 5.216).

When the plunger moves downwards under the action of the hydraulic piston, fuel is pumped through the fuel injector (Fig. 5.216 *a*). This continues until the spool groove on the plunger, connected to the discharge cavity by three vertical channels (Fig. 5.216 *d*), coincides with the discharge hole (Fig. 5.216 *b*). As a result, fuel injection stops until the upper edge of the spool bore closes the unloading hole (Fig. 5.216 *c*), after which injection resumes.

Thus, when the plunger moves downwards, two successive injections occur, the first is short, the value of which is determined by the position of the spool groove on the plunger, and the second is the main one, the value of which is determined by the value of the active stroke of the plunger. The interval between injections depends on the width of the spool bore. The size of the pilot portion in this pump-injector design remains constant in all modes, and the engine is controlled by changing the value of the main supply by the electronic engine control system.

In addition to dividing the cyclic portion into several sequential injections, parallel supply of pre- and main portions of fuel is used in marine diesel engines (Fig. 5.210 *h*).

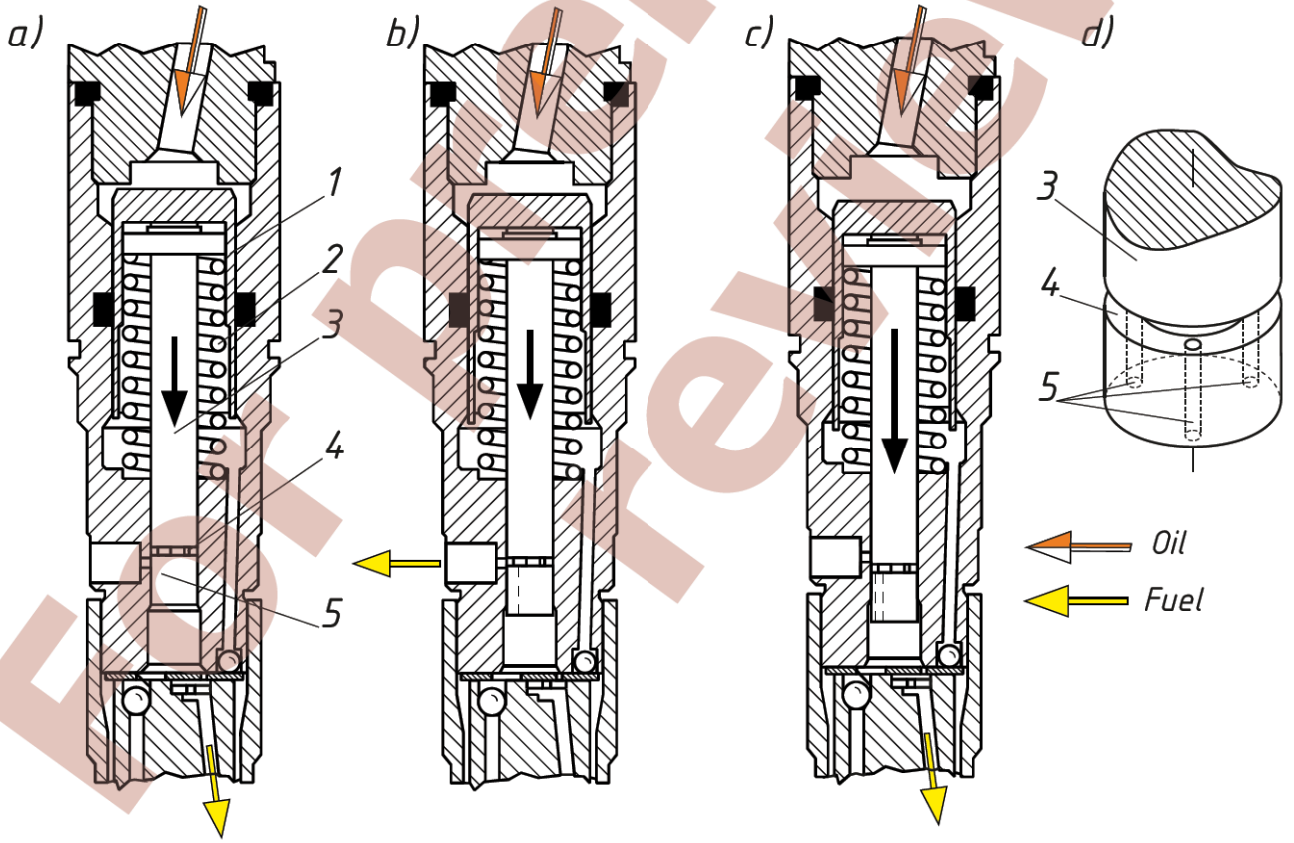


Figure 5.216 – Pump-injector with a hydraulic drive of a Caterpillar 3126 series engine with spool control of two-phase injection: *a* – the beginning of the injection of the pilot portion; *b* – pressure release; *c* – injection of the main portion; *d* – lower part of the plunger with a spool edge; 1 – hydraulic piston; 2 – return spring; 3 – plunger; 4 – spool groove; 5 – fuel bypass connecting channels (adapted from [44, 45])

For preliminary review

This supply split is used in Wärtsilä 46C series engines. To implement this method, two injectors are installed on each engine cylinder, a peripheral one for injection of the pilot portion of fuel and a central one for injection of the main portion (Fig. 5.217).

Both injectors are powered by the same fuel pump, in the cover of which two check valves are installed, adjusted to different opening pressures (Fig. 5.218).

When fuel injection begins, the check valve of the pilot injector opens first, the needle valve of which is adjusted to a lower pressure than that of the main injector. Pilot fuel injection begins.

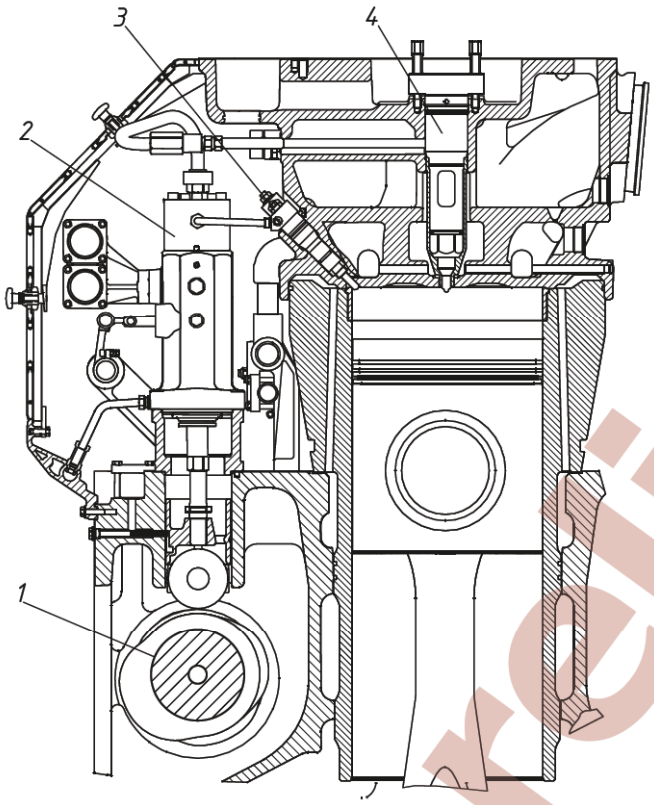


Figure 5.217 – Wärtsilä 46 C series engine injection system with parallel injection of pilot and main fuel portions: 1 – camshaft; 2 – injection pump; 3 – fuel injector for injection of the pilot portion of fuel; 4 – fuel injector for injection of the main portion of fuel (adapted from [119])

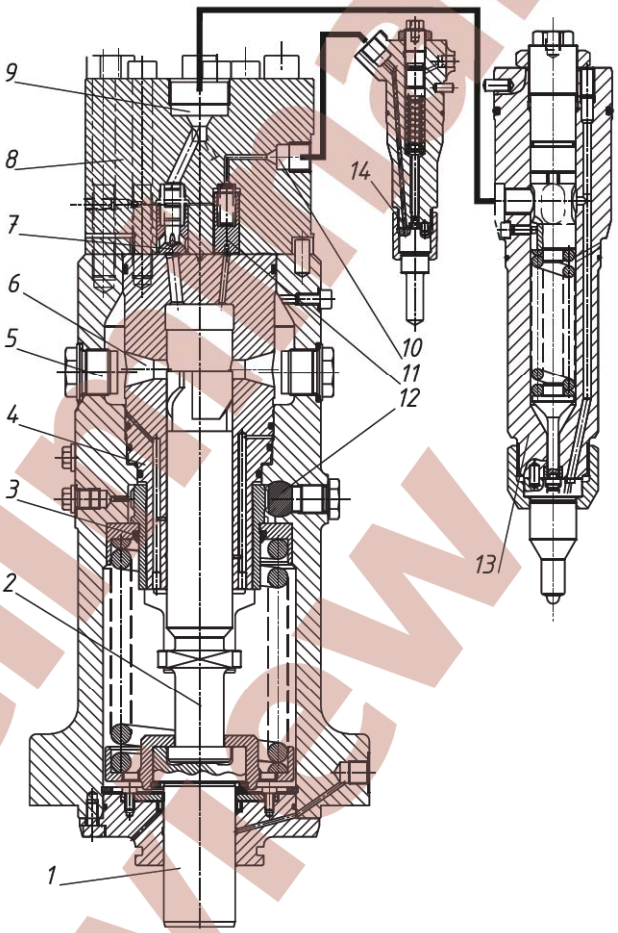


Figure 5.218 – Injection pump of a Wärtsilä 46C series engine with two check valves and sockets for connecting two injectors: 1 – pusher; 2 – plunger; 3 – sleeve for turning the plunger; 4 – plunger bushing; 5 – anti-cavitation plug; 6 – hole for filling the sub-plunger space; 7 – delivery valve; 8 – fuel injection pump cover; 9 – connecting socket of the main fuel injector; 10 – connecting socket of the pilot fuel injector; 11 – check valve of the pilot fuel injector; 12 – rack for turning the plunger; 13 – main supply fuel injector; 14 – pilot fuel injector (adapted from [119])

The parameters of the ignition fuel injector are selected in such a way that by the time the main fuel injector opens, about 10% of the total cyclic portion of fuel appears in the combustion chamber, which by this moment has gone through all stages of pre-flame processes. The main portion is injected into an environment with already formed open flames, which promotes rapid evaporation and combustion of the injected fuel. After the main portion of fuel has been introduced and burned, the ignition fuel injector continues to supply fuel to the combustion chamber for some time, maintaining combustion in it, during which unburned soot particles and unreacted hydrocarbons of the fuel

For preliminary review

burn out. As a result, not only the economic but also the environmental performance of the engine is improved.

The method of parallel and parallel-sequential injection is more widely used in Wärtsilä two-stroke diesel engines of the RT-flex series with a accumulator injection systems. The recent trend towards installing several injectors on one engine cylinder has led to expanded possibilities in organizing the operating process. This forced measure for engines with direct-flow purge schemes turned out to be very convenient for organizing processes with the sequential introduction of individual portions of fuel into the cylinder.

Among the well-known methods of supplying fuel, for example, with three fuel injectors, the following can be distinguished:

- supply of an pilot portion of fuel to one of the fuel injectors, and after the period of time required for the ignition and combustion of the first portion of fuel, the remaining fuel injectors supply the main part of the fuel to the already prepared combustion chamber environment (Fig. 5.219 *a*);

- the first fuel injector supplies the pilot portion of fuel, and after it burns out, the second portion is supplied to the cylinder by the second fuel injector; towards the end of burnout, the third fuel injector supplies the third portion (Fig. 5.219 *b*);

- the injectors supply equal portions of fuel with some delay relative to each other (Fig. 5.219 *c*).

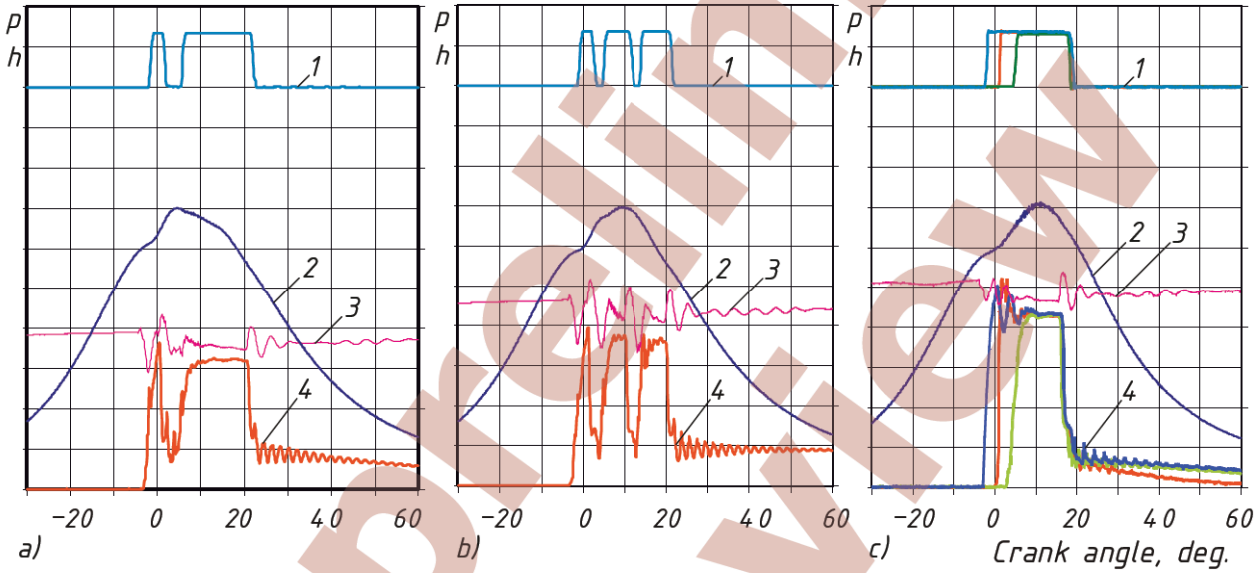


Figure 5.219 – Organization of the operating process and methods of supplying fuel to the cylinder of an engine of the RT-flex series from Wärtsilä with three fuel injectors: 1 – lifting the fuel injector needle; 2 – pressure in the cylinder; 3 – fuel pressure in the accumulator; 4 – fuel pressure in the fuel injector; *a* – sequential supply of the pilot portion of one of the fuel injectors and parallel supply of the main portion by all fuel injectors; *b* – sequential supply by three fuel injectors; *c* – series-parallel supply by fuel injectors with supply delay (adapted from [54])

In all the considered cases, the implementation of the operating process was able to achieve a reduc-tion in fuel consumption and nitrogen oxide emissions.

It should be noted that the widespread introduction of electronically controlled fuel supply systemson marine diesel engines in recent years can significantly improve their efficient and environmental performance. Currently, large-scale research is being conducted aimed at improving fuel supply processes in order to increase the efficiency and environmental performance of marine diesel engines.

Requirements for environmental performance are constantly increasing, and today many manufacturers understand that in the near future it will not be possible to meet these standards only by traditional methods. Today, the search for ways to improve the environmental performance of diesel engines is carried out in three main areas:

- search for new types of fuel, the combustion of which does not produce harmful substances or

For preliminary review

produces them in much smaller quantities. Such fuels include hydrogen, natural and petroleum gases, fuels of biological origin and ammonia;

- development of methods for cleaning engine exhaust gases from harmful substances. Such methods include methods of physical purification from solid particles, chemical purification from harmful chemical compounds;

- development of new approaches to organizing engine operating processes aimed at reducing exhaust gas toxicity.

It should be noted that the problem of increasing environmental performance is usually considered as complex, using all of the listed approaches. Since the first two approaches are beyond the scope of this chapter, it makes sense to dwell only on the third approach. One of the options for increasing the environmental performance of marine diesel engines, aimed primarily at reducing nitrogen oxide emissions, is the use of water in the engine operating process. The next section of the book is devoted to this issue.

5.10 Use of water to improve the environmental performance of marine diesel engines and equipment for supplying water to the engine

The active development of water transport over the past decades has led to a significant increase in its contribution to environmental pollution. This has entailed the intensification of research to develop methods to reduce emissions of harmful substances from the exhaust gases of diesel units, which are one of the main pollutants of the air basin adjacent to navigable waters. The concentration of harmful substances to one degree or another depends on the amount of sulfur in the fuel, its lower calorific value, the type of engine, its speed and indicated efficiency.

Of the entire range of harmful substances, special attention is paid to nitrogen oxides (NO_x), which account for up to 80...90% of the toxicity of exhaust gases. In the human housing, NO_x forms compounds of nitric and nitrous acids, which lead to irritation of the mucous membrane of the eyes and respiratory tract. At a volumetric concentration in the atmosphere of 0.004...0.008% of nitrogen oxides, a person experiences pulmonary edema. Once in the atmosphere, nitrogen oxides participate in photochemical processes with the formation of high concentrations of ozone (O_3) and the appearance of so-called photochemical smog.

The mechanism of formation of nitrogen oxides in internal combustion engines is quite well studied. According to modern understanding, the formation of nitrogen oxides in a diesel engine primarily depends on local combustion temperatures in the combustion chamber [94].

It has long been known that water has a positive effect on the reduction formation of NO_x, reducing temperature peaks during the combustion process. In most cases, this entails a decrease in thermal efficiency and, as a consequence, an increase in fuel consumption. The use of water in the operating process of a marine engine today is considered, first of all, as a means of reducing the toxicity of exhaust gases and only in some cases as a way to increase the efficiency of the diesel operating process. There are three main methods of using water [120]:

- preparation of water-fuel emulsions;
- charge air humidification;
- direct water injection into the working cylinder.

Each of the considered methods has both certain advantages and disadvantages, which ultimately determine the advisability of using a particular method in each specific case.

Considering that the use of water-fuel emulsions is beyond the scope of the format of this book, we will dwell in more detail on the methods of humidifying the air entering the working cylinders and direct injection of water into the working space of the engine. Since water in all the cases considered is involved as the third component of the mixture (water-fuel-air), we classify the first method as external mixture formation, and the second as internal.

5.10.1 Methods for saturating incoming air with water vapor

MAN company for NO_x emissions reduction investigated the possibility of charge air humidification. For this purpose, the German company Munters Euroform has developed a system called Hu-

For preliminary review

mid Air Motor (HAM), which allows you to increase air humidity to 99% (Fig. 5.220).

Air compressed and heated to 120...130°C from the turbocharger passes through a humidifying column, in which it is cooled and humidified. In this case, the relative humidity of the air entering the engine can be maintained at a constant level of 99%.

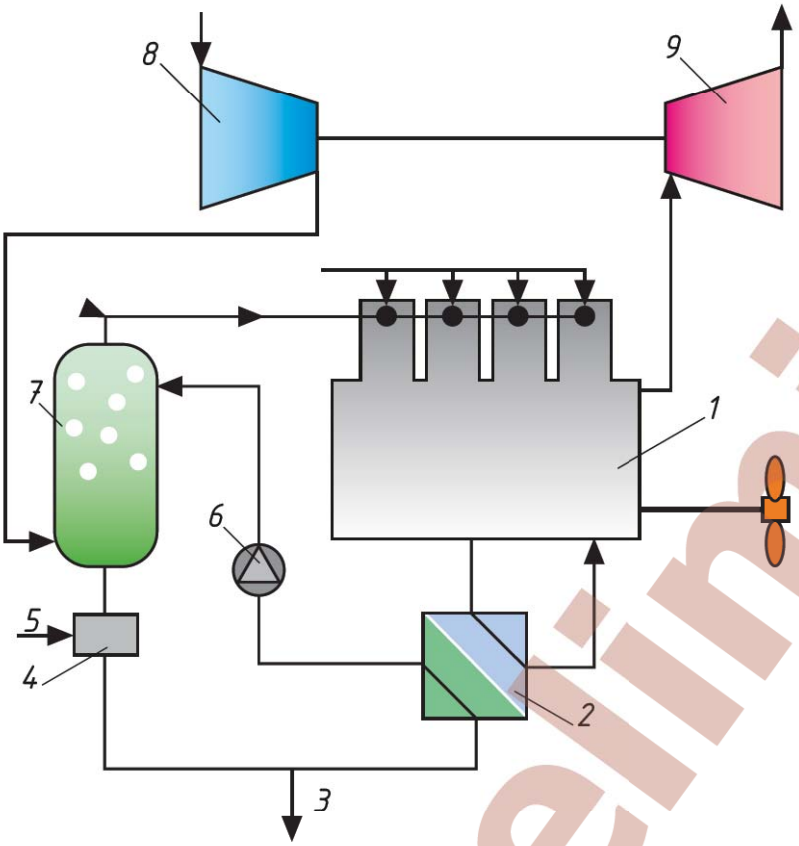


Figure 5.220 – Diagram of the installation for air humidification at the inlet to the engine from Munters Euroform: 1 – engine; 2 – heat exchanger; 3 – water drain; 4 – water tank; 5 – water supply; 6 – circulation pump; 7 – humidifying column; 8 – supercharging compressor; 9 – turbocharger turbine (adapted from [121])

Testing of the system showed that in operating mode the NO_x content decreased by 70...80%. Researchers explain this by the fact that the increased vapor content in charge air reduces temperature peaks in the combustion chamber.

Wärtsilä company in its early developments used a device for saturating the air entering the combustion zone with water vapor (Combustion Air Saturation System (CASS)) from Marioff Oy company. A water aerosol was introduced through nozzles directly into the charge air flow immediately after the turbocharger (Fig. 5.221).

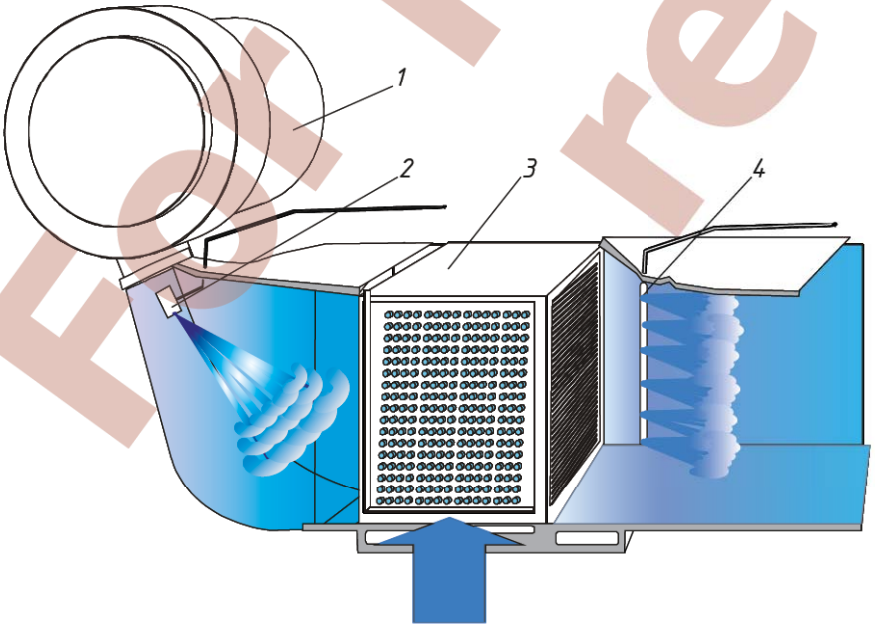


Figure 5.221 – Wärtsilä air humidification installation: 1 – turbocharger; 2 – injector for water injection; 3 – air cooler; 4 – additional water injection (adapted from [122])

For preliminary review

Droplets, several microns in size, evaporate very quickly in an environment with a temperature of more than 473 K and a flow speed of 75 m/s. The remaining droplets evaporated in an air cooler, which acted as a heater. Every kilogram of into the combustion zone air satiated 60 gram water. How should from reports from Wärtsilä, this ensured a reduction NO_x up to 3 g/(kWh). In case of insuffi-cient humidity, the possibility of additional spraying of a water aerosol after the air cooler was con-sidered.

5.10.2 Methods of direct water injection into the engine working space

Water can be introduced into the combustion chamber through separate nozzle tips mounted in one fuel injector, through a separate fuel injector installed in the cylinder cover, or by stratified injection of water and fuel through the nozzle tip of one fuel injector [123].

Wärtsilä company in its developments it followed the path of direct injection of water into the combustion chamber of serial engines. The volume of supplied water is 15...70% of the cyclic fuel supply. For water injection, a fuel injector with two fuel tips (Fig. 5.222) for separate fuel and water supply, as well as a water supply system to the fuel injector (Fig. 5.223).

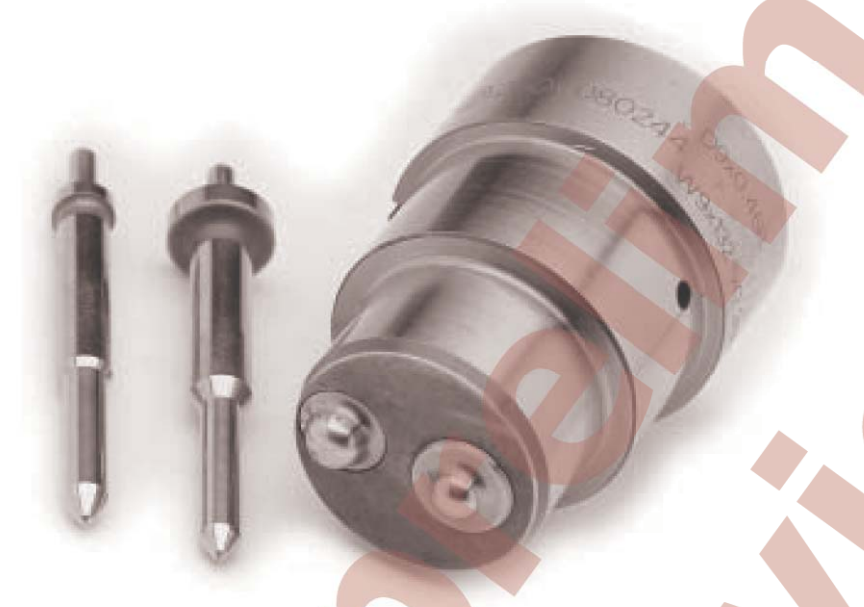


Figure 5.222 – Nozzle for joint injection of water and fuel into the combustion chamber of a 46 series engine from Wärtsilä (adapted from [122])

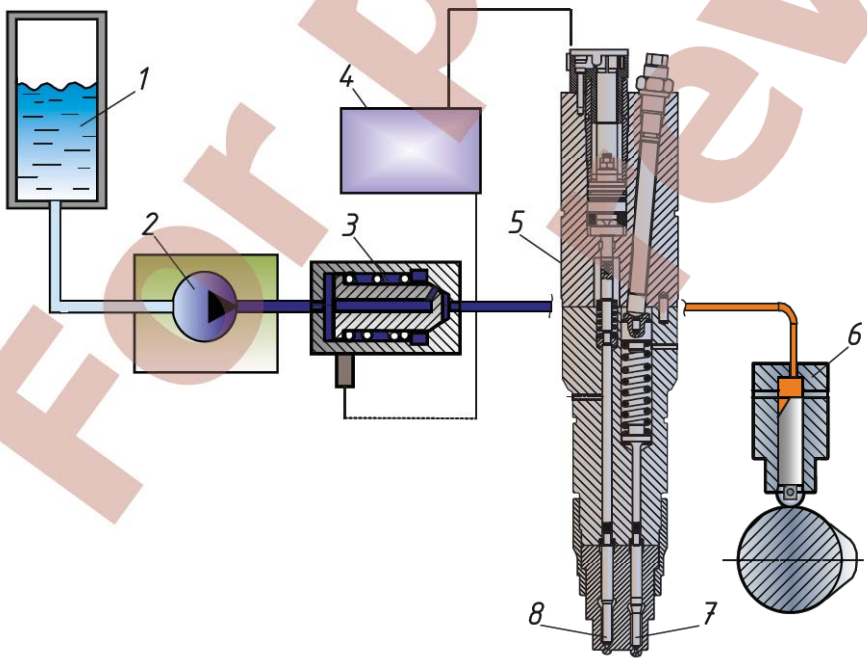


Figure 5.223 – Diagram of the water injection system into the combustion chamber through a fuel injector with a double nozzle tips: 1 – water tank; 2 – pressure water supply pump; 3 – maximum water supply limiter; 4 – electronic control unit; 5 – fuel injector; 6 – injection pump used to supply liquid fuel; 7 – tip for spraying liquid fuel; 8 – tip for spraying water (adapted from [124])

For preliminary review

For water injection, a high pressure fuel pump is used to supply water to the accumulator. Water is supplied to the fuel injector through a maximum flow limiter at a pressure of 21 MPa.

The electronic control system allows you to change the water supply depending on the engine operating mode within a wide range and provide an emergency shutdown of its supply in the event of a stuck spray needle. Tests of the system have shown that its use makes it possible to reduce NO_x emissions by 50...60% without a noticeable reduction in power.

The system is now offered by Wärtsilä as an additional option for new engines and for retrofitting existing engines that are equipped with single center injector fuel systems.

a accumulator system for separate injection of water and fuel with electronic control of water supply was developed. The layout of the main elements of this system is shown in Fig. 5.224.

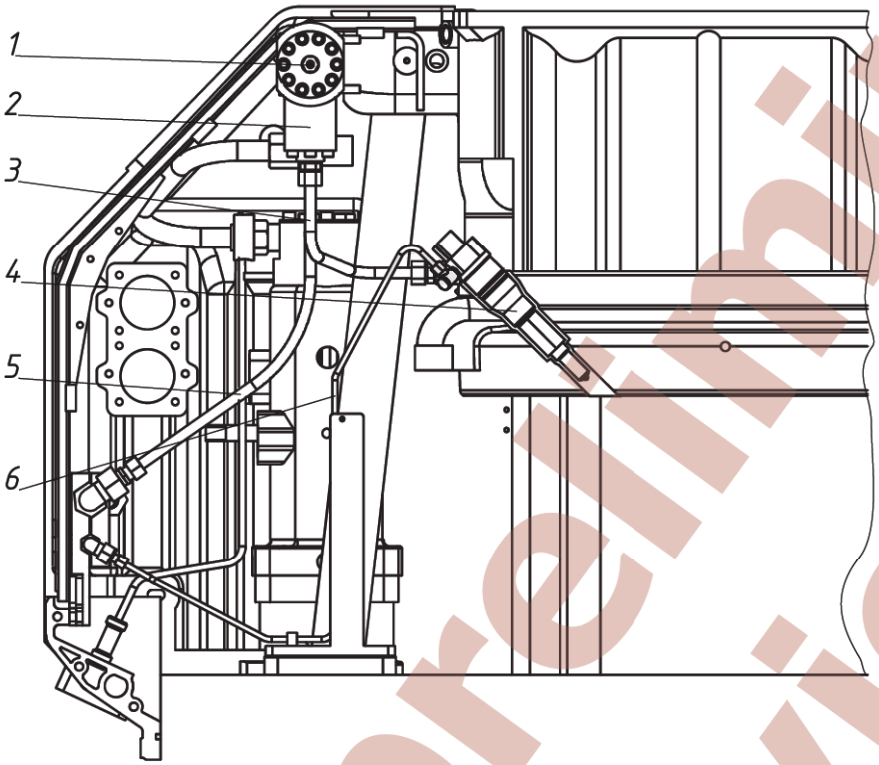


Figure 5.224 – Location on the engine of the elements of the water injection system into the combustion chamber of the Wärtsilä 46 series engine: 1 – water accumulator; 2 – accumulator input unit; 3 – high pressure pipe for supplying water to the injector; 4 – water injector; 5 – high pressure pipe for supplying water to the accumulator; 6 – oil line for injector control (adapted from [125])

For water injection, a injector installed peripherally in the cover of the slave cylinder was used. Liquid fuel injection was carried out by the standard engine fuel system through a fuel injec-tor installed along the cylinder axis.

In 1999 the company equipped with separate injection systemswater seven ro-ro vessels for Transfennica. The vessels with a deadweight of 7100...7250 tons were built at the German shipyard JJ Sietas and equipped with 12- and 16-cylinder 46 series engines from Wärtsilä.

More than twenty years old operation of ships has shown that this technology has made it possible to reduce NO_x emissions by 50...60%, and the loss of efficiency did not exceed 2...3 g/(kWh). The nature of the operating process of an engine with water injection is shown in Fig. 5.225.

As can be seen from Fig. 5.225, the water injection begins 45° before TDC and ends immediately before the start of fuel injection 10° before TDC.

The evaporation of a water aerosol is accompanied by a decrease in temperature in the combustion chamber, and the presence of a large amount of water vapor in the charge leads to an increase in its heat capacity. The combined action of these two factors leads to a decrease in temperatures in local combustion sites and a reduction in the formation of nitrogen oxides.

Direct water injection provided the optimal combination of good economy and reduced exhaust emissions. This is achieved with a one-time, relatively low investment, and a slight increase in operating costs without compromising reliability and, in fact, without additional requirements for equipment placement.

For preliminary review

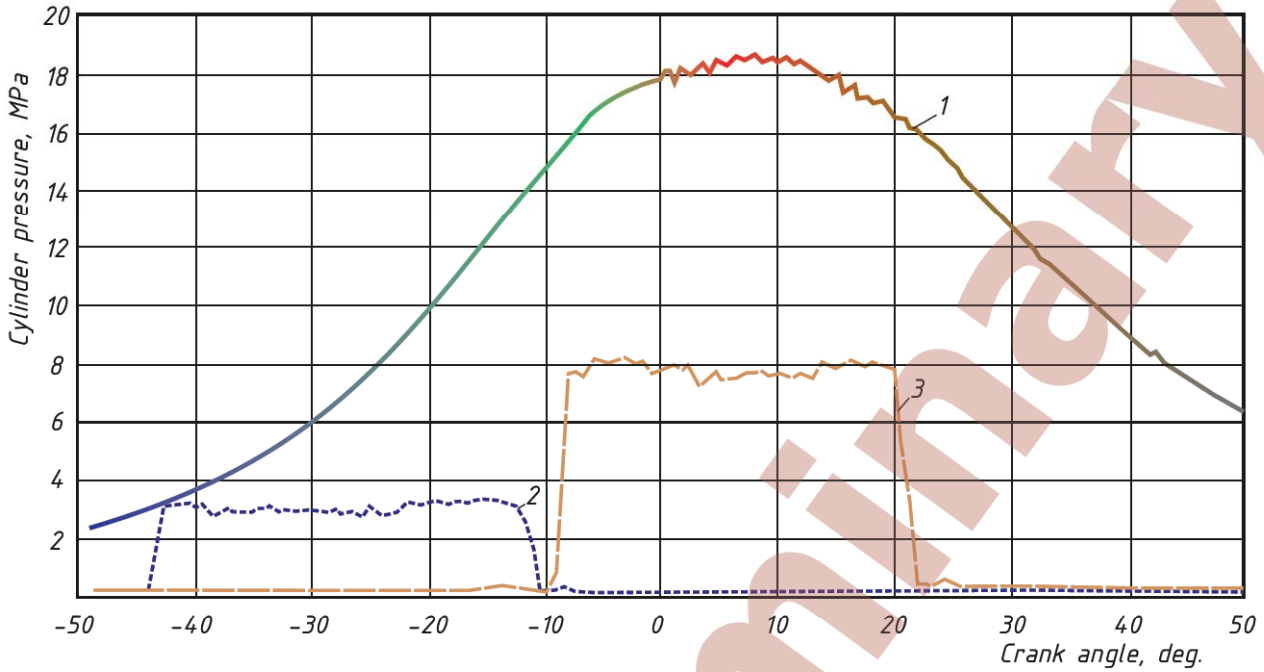


Figure 5.225 – The nature of the operating process in the 46 series engine from Wärtsilä with separate injection of water and fuel into the combustion chamber: 1 – pressure in the cylinder; 2 – raising the needle of the water injector; 3 – lifting the fuel injector spray needle (adapted from [124])

Mitsubishi Heavy Industries researched stratified fuel-water injection (SFWI) via one spray, equipped special valve device, allowing during periods be-tween injections fill water Part internal cavities high pressure pipeline (Fig. 5.226 *a*). Thus, clearly defined zones of alternation of water and fuel were formed in the fuel flow supplied to the fuel in-jector.

When injecting sequentially were forced out fuel, water and again fuel, forming in atomization plume spraying clearly expressed layers (Fig. 5.226 *b*). Device provided stable work low-speed Mitsubishi UEC 52/105 engine in all operating modes. Noted reduction emissions NO_x was directly proportional quantity introduced water.

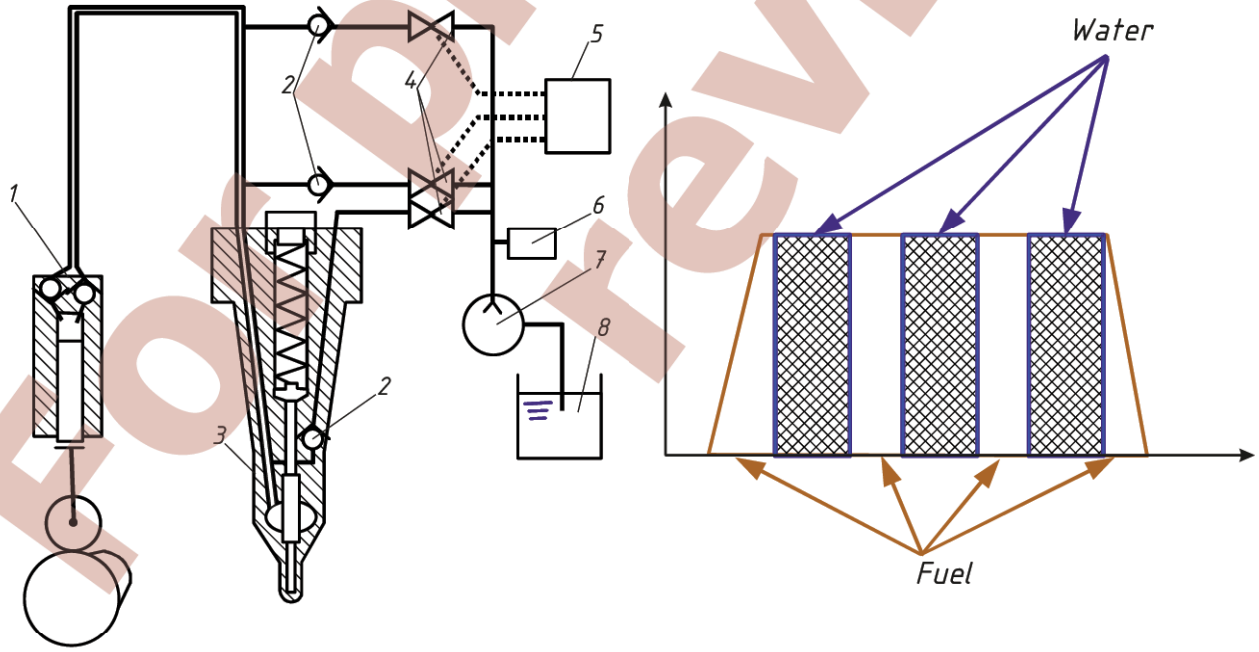


Figure 5.226 – Water injection system of low-speed diesel engine UEC 52/105 from Mitsubishi: 1 – injection pump; 2 – check valves for supplying water to the fuel injector and high pressure line; 3 – fuel injector; 4 – electrically controlled locking devices; 5 – electronic control unit; 6 – pressure accumulator; 7 – booster pump; 8 – water tank (adapted from [126])

For preliminary review

For high-speed engines, stratified systemsare considered as promising injection fuel And water through one spray, equipped special spool valve device, allowing during periods between injections fill water Part internal cavities, adjacent To fuel injector needle. The diagram of such a nozzle is shown in Fig. 5.227.

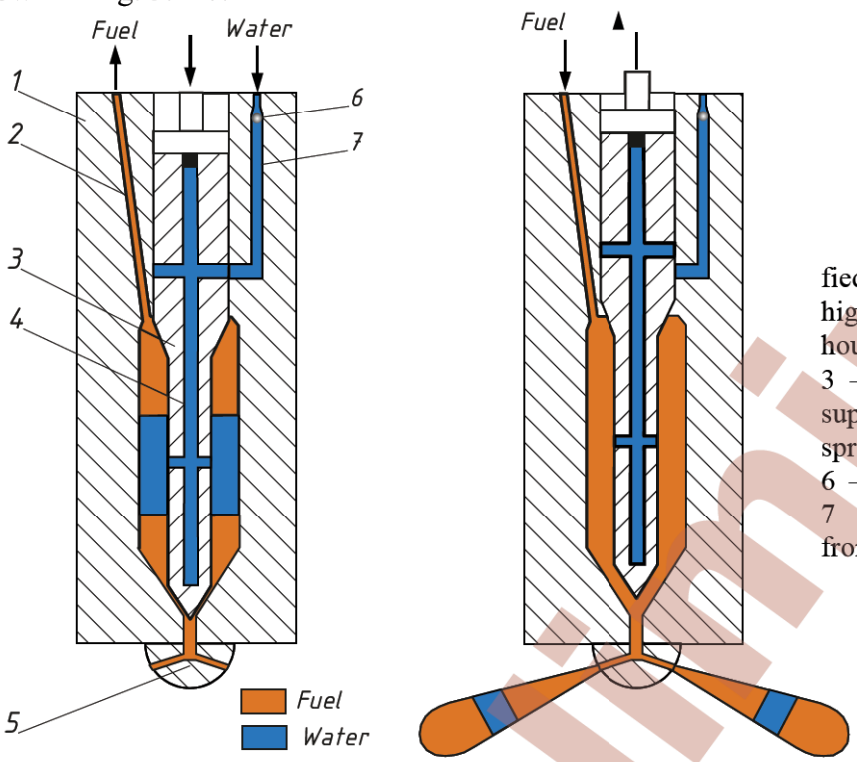


Figure 5.227 – Nozzle for stratified injection of water and fuel in a high-speed diesel engine: 1 – nozzle housing; 2 – fuel supply channel; 3 – needle valve; 4 – channel for supplying water to the cavity of the sprayer; 5 – fuel nozzle tip; 6 – check valve of the water line; 7 – water supply channel (adapted from [127])

When the needle valve is closed, the water supply channel in the nozzle housing coincides with the channel in the housing of the needle valve. The pressure of the supplied water is slightly higher than the residual pressure in the fuel line, so part of the water enters the the cavity of the sprayer, displacing the fuel and forming a water layer in the the cavity of the sprayer. When the fuel injection pump is pumping fuel, the water supply channel is closed by a check valve, and when the needle valve is raised, the water and fuel cavities are additionally disconnected due to the water supply channel being blocked in the nozzle housing of the needle valve generatrix.

As a result, clearly defined zones of fuel and water are formed in the atomization plume formed by the fuel injector.

Literature used when writing section 5:

1. Vermeire M.B. Marine products everything you need to know about marine fuels. Chevron Marine Products. Ghent, Belgium. 2021 – 26 p.

2. Research of the Gas Fuel Supply Process on the Compression Stroke in Ship's Low-Speed Gas-Diesel Engines / Belousov E. Marchenko A., Gritsuk I., Savchuk V., et al. // SAE Technical Paper 2020-01-2107, 2020, doi:10.4271/2020-01-2107.

3. Белоусов Е.В. Организация внутреннего смесеобразования в судовых малооборотных газодизельных двигателях. / Е.В. Белоусов, В.П. Савчук, И.В. Грицук, Т.П. Белоусова: Двигатели внутреннего сгорания // Научно-технический журнал. Харьков: НТУ (ХПИ). – 2017. – №2. – 68 с., с. 13-16.

4. Gas as a marine fuel contractual guidelines. Society for Gas as a Marine Fuel, 2015. – 13 p.

5. LPG for Marine Engines The Marine Alternative Fuel Commercial, Passenger, Offshore Boats/Ships, Recreational Crafts and Other Boats. Innovation & Technology. Neuilly-sur-Seine, France. 2021. – 150 p.

6. DeLuca F. History of fuel injection. Bosch Arma Corporation. – 22 p.

7. Белоусов Е.В., Billier P. Французский инженер Sabathe Louis Gaston и его роль в

For preliminary review

создании двигателей со смешанным подводом теплоты. Двигун внутрішнього згорання. Всеукраїнський науково-технічний журнал. – Харків: вид. НТУ «ХПІ» – 2021. – № 1. – С 92-98.
8. Wärtsilä 26 Technology Review. Wärtsilä Corporation. Helsinki, Finland. 2008. – 8 p.
9. Белоусов Е.В. Топливные системы современных судовых дизелей: учебное пособие – Херсон: ХГМА, 2014. – 268 с.
10. Technical Documentation Engine Operating Instructions L32/40CR. 6709 B1-4 EN, MAN Diesel SE, Augsburg. – 352 p.
11. Engine type W46B Instruction Manual. Wartsila Finland Oy Vaasa Factory. Finland. 2011. – 458 p.
12. Maintenance 8M 25 C. Caterpillar Motoren GmbH & Co. KG., Kiel. 2014. – 381 p.
13. Project Planning Manual for Marine Main Engines V 40/50. S.E.M.T. Pielstick's, MAN B&W Diesel AG's. – 166 p.
14. Operating Instructions TBD 645. 0299 6391 en, Motoren-Werke Mannheim AG, Mannheim. Germany. – 244 p.
15. Service Bulletin. Technical Information to all the Owners of Sulzer S 20 Type Diesel Engines New Uncooled Fuel Injection Nozzle for MDO Operation. New Sulzer Diesel Ltd.Winterthur, Switzerland. – 4 p.
16. Sulzer RTA58T Operating Manual «Marine». Wärtsilä NSD Switzerland Ltd. Winterthur. 1998 – 999 p.
17. Instructions for Main Engine Type S50MC-C. Vol. III Components And Descriptions. MAN B&W Diesel A/S. Copenhagen, Denmark. – 168 p.
18. Hitachi-Wärtsilä RT-flex50-B. Spare Parts Code Book «Marine». Hitachi Zosen Corporation. Osaka, Japan. – 296 p.
19. Service Bulletin. Technical Information to all the Owners of Sulzer RTA 38. 48 and RTA -2 and -2U Series Engines with Oil-Cooled Pistons. Wärtsilä NSD Switzerland Ltd. Winterthur, Switzerland. 1998. – 9 p.
20. Engine Type S50ME-C8. Volume III Components Description. DS 42014238. STX Dalian Engine Co.,Ltd. – 202 p.
21. Service Bulletin. Technical Information to all the Owners of Sulzer RTA – Type Diesel Engines except RTA38/48/52U/62U/72U and 84T. Circulation Valve to Fuel Injection Valve. Wärtsilä NSD Switzerland Ltd. Winterthur, Switzerland. 1995. – 9 p.
22. Container Ship Propulsion. MAN B&W Diesel. 1997. – 16 p.
23. Fuel Injection MAN B&W L27/38. Pipe 51404-08H. Holeby Generatinc Sets, 2012. – 2 p.
24. Engine operating instructions. Type M32C. Caterpillar Motoren GmbH & Co. KG. 2007. – 2632 p.
25. RTA72U-B. Operating Manual «Marine». Wärtsilä Switzerland Ltd, Switzerland. 2000. – 277 p.
26. Instructions for Main Engine Type 10K98MC-C. Vol. III Components and descriptions. HSD engine Co., Ltd. Kyungnam, Korea. – 212 p.
27. MAN B&W L16/24. Fuel Injection Pump (L'Orange). 51401-07H, Edition 07H., 08028-0D/H5250. Holeby Generatinc Sets, 1994. – 3 p.
28. Instruction Book «Operation» for 50-108MC/MC-C Engines. General Edition 0002. STX Engine Co.,Ltd. Kyungnam, Republic of Korea. 2004. – 1155 p.
29. Rafalski R. Wärtsilä 46F optimized with DocHub e-932 WLAN. Engine Monitoring Report USA, Logistic & Service Center International Technology GmbH, January 2014. – 5 p.
30. Technology review Wärtsilä 46F engine. Wärtsilä Corporation. Helsinki. 2004 – 20 p.
31. Goering, Carroll E., Marvin L Stone, David W. Smith, Paul K. Turnquist. Compression-ignition engines. Chapter 7 in Off-Road Vehicle Engineering Principles, . St. Joseph, Mich.: ASAE. 2003. – P. 131-162.
32. Service Bulletin. Technical Information to all Owners / Operators of Sulzer RTA Engines (except RTA 36 and RTA 48). Variable Injection Timing and Fuel Quality Settingl. Wartsila Switzerland Ltd. Winterthur, Switzerland. – 17 p.

For preliminary review

33. Sulzer RTA48T Operating Manual «Marine». Wärtsilä NSD Switzerland Ltd. Winterthur, Switzerland. 2000. – 294 p.
34. Replacement Parts for Caterpillar. 3500 Series Engines. Interstate-Mcbee, LLC. Cleveland, Ohio, USA. – 77 p.
35. Signature and ISX CM870 Fuel System Cummins Ontario Training Center. Cummins Ontario Inc. Mississauga. – 16 p.
36. The Intelligent Engine: Development Status and Prospects. – 19 p.
37. Environmental Friendly Diesel Engine «UEC Eco-Engine» Sakaguchi K., Yoshikawa S., Sugihara M., Edo K., Ito K., Murata S., Tanaka A., Sonoda T. Mitsubishi Heavy Industries, Ltd. Technical Review Vol.41 No.1. 2004 – 3 p.
38. Service Experience 2008, MAN B&W Engines ME/ME-C and MC/MC-C Engine Series. MAN Diesel, Copenhagen, Denmark. 2008. – 20 p.
39. Service Experience 2007, MAN B&W Engines The ME/ME-C and MC/MC-C Series. MAN Diesel A/S, Copenhagen, Denmark. 2007. – 29 p.
40. Hydraulic Power Supply. MAN Diesel A/S. 2006. – 39 p.
41. MAN Diesel K98ME. Volume III – Components Description. DS 09092965. MAN Diesel A/S. – 219 p.
42. Flarup J. Fuel Equipment and Hydraulics – New Developments. Paper No. 2 – 2014 – 20 p.
43. Hagen P. ME Technology – Further Improvements, Fuel injection, Copenhagen June 2012. Extract from Licence Days. MAN Diesel & Turbo. 2012. – 13. p.
44. Hydraulically Actuated Electronic Pump-injector (HEUI) Systems. Caterpillar, IAP Control Circuit, Pressures. 2001 – 42 p.
45. Machine Electronics. Lesson 3: Hydraulic Electronic Unit Injection (HEUI). – 40 p.
46. Systems Operation 3126B and 3126E Truck Engine. RENR1271-11. Caterpillar, 2006. – 25 p.
47. HEUI HI300B Component Operation High Efficiency Pump. HEUI B System Features. Caterpillar, 2001– 14 p.
48. HEUI HI300B Component Operation High Efficiency Pump. HEP Operation. Caterpillar, 2001 – 10 p.
49. Sulzer RT-flex low-speed engines. W0304E Wärtsilä Corporation. Helsinki. 2004. – 9 p.
50. RT-flex Training. Function of the different Hydraulic- and Pneumatic Components. Chapter 30. Wärtsilä Land & Sea Academy. 2007. – 21 p.
51. Wärtsilä RT–flex96C Operating Manual «Marine» (WECS–9520 / Dynex Servo Oil Pumps). Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2007 – 373 p.
52. Wärtsilä RT-flex50-D Operating Manual. «Marine» (with Pulse Lubrication). Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2010 – 502 p.
53. Wärtsilä RT-flex50-D. Ersatzteil-Kodebuch Spare Parts Code Book «Mahne». Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2012 – 374 p.
54. Heim K., Brown D. State-of-the-Art Common-Rail Technology in Low-Speed Marine Diesel Engines. Wärtsilä Corporation. Winterthur, Switzerland. 2009. – 18 p.
55. RT-flex Training. Function of the different Hydraulic- and Pneumatic Components. Chapter 30. Wärtsilä Land & Sea Academy. Wärtsilä Corporation. 2007. – 21 p.
56. RT-flex Training. Mechanical Features (RT-flex Specific Parts only). Chapter 20. Wärtsilä Land & Sea Academy. Wärtsilä Corporation. 2007. – 43 p.
57. Wärtsilä W-X35. Maintenance Manual «Marine». Document ID: DBAC352248. Winterthur Gas & Diesel Ltd. Winterthur, Switzerland. 2015. – 488 p.
58. Service experience. MAN Energy Solutions. Future in the making. MAN Energy Solutions, Denmark. 2019 – 52 p.
59. Vogel C., Wachtmeister G., Maier L., Haas S., Marzinek A. New concept of HFO common rail injection system for MAN B&W MS-Diesel engines. Paper No.: 136. Kyoto, CIMAC Congress 2004. – 11 p.
60. Technical Documentation Engine. Operating Instructions. Engine L32/44CR. MAN Diesel

For preliminary review

SE. Augsburg. 2008. – 356 p.
61. Technical Documentation Engine. Working Instructions. Engine L32/44CR. MAN Diesel SE. Augsburg. 2008. – 875 p.
62. Common Rail Design and maturity. MAN Diesel & Turbo. Augsburg, Germany. – 18 p.
63. Emission control. The common rail system. Wärtsilä Finland Oy. Vaasa, Finland. – 4 p.
64. Wärtsila W12V32, Instruction Manual. Wärtsila Finland Oy. Vaasa, Finland. – 462 p.
65. Fuel System Wärtsilä L46 Common Rail. W4602L00BTM04A Rev. 03. Wärtsilä. 2007. – 61 p.
66. Wärtsilä 46 Common Rail. Wärtsilä. – 60 p.
67. Automation system. Wärtsilä V46 WECS 2000 & WECS 7500 Common Rail. W4608V00BTM10A Rev. 01. Wärtsilä. 2007. – 40 p.
68. Wärtsilä 46 Common Rail. Wärtsilä. 2001. – 40 p.
69. Caterpillar Common Rail. Less Fuel and Reduced Emissions Mean More Environmental Care. Caterpillar Marine Power Systems. Hamburg, Germany. 2008 – 2 p.
70. Engine 125E-5 Series. Structure, function and maintenance standard, Part 2. Komatsu, Japan. 2005 – 58 p.
71. Diesel Common Rail Systems for Passenger Cars. Powertrain Control. – 2 p.
72. New Holland LB90.B-LB115.B Engines F4GE0454C-F4GE0484G 604.13.612 Repair Manual. Italy. –117 p.
73. From high speed diesel to medium speed HFO engines – L'Orange platform based common rail pumps fulfill future demands. / Jorach R.W., Schneider H., Kerckhoff A.J., Altmann O. // Paper No.: 80, CIMAC Congress, Kyoto. 2004. – 14 p.
74. Common rail system (CRS). Service manual: General Edition. Denso Corporation. 2008 – 168 p.
75. Common rail manual. Principles of operation. DDGX200(EN). Delphi Diesel Systems Ltd. 2007 – 83 p.
76. Service Manual. Common Rail System for Isuzu 6HK1/6SD1 Type Engine. Operation. Denso corporation. 2003 – 23 p.
77. TDI engine with a displacement of 2.8 liters and a Common Rail injection system. Device and principle of operation. Volkswagen AG, Wolfsburg. 2003 – 56 p.
78. Kitchen T. Common Rail Diesel Fuel Systems. AK Training. – 61 p.
79. Technician guide. Series 4000 Common Rail Fuel System. Detroit Diesel Corporation. 2001 – 134.
80. Soruşbay C. Internal Combustion Engines. Mixture preperation in CI engines. – 6 p.
81. Technical Documentation Diesel Engine 12 V 4000 P63, P83; 16 V 4000 P63, P83; 20 V 4000 P63, P83. Application Group 3B. Operating Instructions. MS150050/00E. MTU Friedrichshafen GmbH, Germany. – 154 p.
82. Freiberger S., Nagel A. Understanding the communication between automotive mechatronics and electronics for remanufacturing purposes. European Automotive Remanufacturing Technical Trends & Market Development. Bayreuth University. 2012 – P.15-42.
83. Анализ современных подходов к проблеме создания судовых малооборотных газодизельных двигателей. / Е.В. Белоусов, В.П. Савчук, Т.П. Белоусова: Двигатели внутреннего сгорания. Всеукраинский научно-технический журнал. – Харьков: изд. НТУ(ХПИ) – 2016. – № 1. – С 81-88.
84. Engine technology 12 and 18V34SG. Wärtsilä Finland Oy. Vaasa, Finland. 2005 – 16 p.
85. Stenersen D. Gas Fuelled ships LNG-Fuelled Engines and Fuel Systems for Medium- Speed Engines in Maritime Applications. Marintek GTS Technical Seminar Series, 28.09.2011. – 38 p.
86. Caterpillar Gas Engines Application and Installation Guide G3600–G3300. Caterpillar Inc. USA. 1997 – 50 p.
87. Thijssen B. Dual-fuel-electric LNG carriers. LNG Shipping Operations. Wärtsilä Finland Oy. Ship Power Solutions. Hamburg, September 27, 2006 – 49 p.
88. Giernalczyk M., Silowni Okretowych K. Analysis Of Possibilities Of The Application Of

For preliminary review

Dual Fuel Engines As A Main Propulsion On Lng Gas Carriers. Journal of KONES Powertrain and Transport, Vol. 15, No.4, 2008 – P. 147-155.

89. Wärtsilä W32DF. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 256 p.

90. Grant J. 50DF Gas Engines in LNG – Development. Wartsila UK Ship Power. Wärtsilä, Presentation 08 September 2009. – 79 p.

91. Определение углов открытия и закрытия газовых клапанов при организации внутреннего смесеобразования в судовых малооборотных газодизельных двигателях. / Е.В. Белоусов, И.В. Грицук, В.С. Вербовський: Двигатели внутреннего сгорания. Харьков: изд. НТУ(ХПИ) – 2018. – № 2. – С 21-25.

92. Исследование процессов топливоподачи в газодизельных малооборотных двухтактных двигателях низького давления. / Варбанец Р.А, Савчук В.П., Грицук И.В., Вербовский В.С. // Двигатели внутреннего сгорания. Харьков: изд. НТУ«ХПИ» – 2019. – № 1. – С 27-33.

93. Исследование влияния давления подачи и сечений сопловых каналов на процессы топливоподачи в газодизельных малооборотных двухтактных двигателях низкого давления. / Марченко А.П., Савчук В.П., Вербовский В.С. // Двигатели внутреннего сгорания. Харьков: изд. НТУ «ХПИ» – 2020. – № 1. – С 27-33.

94. Білоусов Є.В. Теоретичні основи робочих процесів в суднових двигунах, що працюють на альтернативних паливах: монографія / Є.В. Білоусов. – Херсон: Олді-Плюс, 2020. – 444 с.

95. Ott M. WinGD low-speed engines. X-DF low-pressure dual-fuel engine technology. Licensees Conference 2015. – 7 p.

96. The 2-stroke Low-Pressure Dual-Fuel Technology: From Concept to Reality. 02 Product Development – Gas & Dual Fuel Engines. / Nylund I., Alder R., Hirose T., Umemoto Y., Yamada T. // Paper No. 233, CIMAC Congress, Helsinki 2016 – 12 p.

97. Future in the making MAN B&W ME-GA. Propulsion engine. MAN Energy Solutions, Copenhagen, Denmark. 2021 – 2 p.

98. Ott M. Wärtsilä 2-Stroke Dual Fuel Technology. Dual fuel technology update. Technical Seminar 2013 – 32 p.

99. X-DF Engines. Excellence built in Low-pressure X-DF Engines FAQ. Winterthur Gas & Diesel Ltd. 2018 – 20 p.

100. Aabo K. Høytrykks saktegående totakts gassmotor for større skip. Den Norske Gasskonferansen LNG operation 2 stroke and 4 stroke Genset Marine Engines. 27. of March 2014. – 66 p.

101. ME-GI – Dual Fuel Done Right. MAN Diesel, ME-GI, Sname NY, Dec 2013 – 73.

102. ME-GI Dual Fuel MAN B&W Engines. A Technical, Operational and Cost-effective Solution for Ships Fuelled by Gas. MAN Diesel & Turbo. Copenhagen, Denmark. 2012. – 36 p.

103. LNG Carriers with ME-GI Engine and High Pressure Gas Supply System. MAN Diesel A/S, Copenhagen, Denmark. – 37 p.

104. Winkel O. LNG for deep sea operation. Low Speed – Promotion & Sales. MAN Diesel & Turbo. 2012 – 30 p.

105. The 4T50ME-GI Test Engine at MAN Diesel & Turbo's Diesel Research Centre Copenhagen. MAN Diesel A/S, Copenhagen, Denmark. 2011 – 21 p.

106. Jensen J. Retrofit: MAN B&W two stroke engine. MAN Diesel & Turbo, Copenhagen. 2013. – 43 p.

107. Ott M. Wärtsilä 2-Stroke Dual Fuel. Technology responding to changing market needs. Wärtsilä. 5TH CIMAC CASCADES, Busan. 2014 – 25 p.

108.

109. Utilisation of Volatile Organic Compoundsin Shuttle Tankers and Unrefined oil Carriers. MAN Diesel. – 24 p.

110. Белоусов Е.В. Создание и совершенствование твердотопливных поршневых двигателей внутреннего сгорания. – Херсон: ОАО «ХГТ» – 2006. – 452 с.

111. Injection Spray Analyses From Medium Speed Engines Using Marine Fuels. / Fink C., Buchholz B., Niendorf M., Hamdorf H. // Paper ID ILASS08-6-9. ASS, Como Lake. Italy. 2008 – 9 p.

For preliminary review

112. Hengeveld J., Schenk C., Aabo K. The role of temperature and pressure in the wear processes in low speed diesel engines. Shell Global Solutions International, Amsterdam, Netherlands. – 7 p.
113. Bilousov, I., Bulgakov, M. & Savchuk, V. Modern Marine Internal Combustion Engines. Springer International Publishing. 2020. – 385 p.
114. Eisen S., Ofner B., Mayinger F. Investigations Of Common-Rail Fuel Injection Technique In DI-Diesel-Engines. Technische Universit at Munchen, Germany. – 14 p.
115. LNG Carrier Propulsion by ME-GI Engines and/or Reliquefaction. MAN B&W Diesel A/S, Copenhagen, Denmark. – 26 p.
116. Belousov, E., Marchenko, A., Rybalchenko, M., Tuluchenko, G. et al., Digital Profiles of Work Processes of a Marine Engine for Calculated Indicator Diagrams in Operating Modes Other than Maximum Continuous Rating, SAE Technical Paper 2023-01-5008, 2023 – 13 p.
117. M32C Low Emission Engine. Caterpillar Marine Power Systems. Hamburg, Germany. 2009 – 12 p.
118. Technical Documentation Engine Spare Parts Catalogue. Engine 9L 32/40. MAN B&W Diesel AG. 1996 – 248 p.
119. Wärtsilä 46 C. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 421 p.
120. Белоусов Е.В., Агеев М.С., Свиридов В.И. Влияние на рабочий процесс среднеоборотного судового дизеля путем впрыска воды в рабочий цилиндр. Двигатели внутреннего сгорания. – Харьков: изд. НТУ«ХПИ» – 2010. – № 1. –С 40-43.
121. Nord K. Particles and Unregulated Emissions from CI Engines Subjected to Emission Control. Luleå University of Technology, Luleå, Sweden. 2005 – 197 p.
122. The EnviroEngine Concept. Wärtsilä Corporation. 2004 – 8 p.
123. Белоусов Е.В. Моделирование процесса сжатия с охлаждением воздушного заряда путем распыливания воды в рабочем цилиндре ДВС // Двигатели внутреннего сгорания. 2006. – № 1. – С. 72-78.
124. Steffens D. The Diesel Engine and the Environment. Wärtsilä North America, Inc. Houston, Texas. September 16-17, 2003 – 36 p.
125. Wärtsilä 46 Technology Review. Wärtsilä Corporation. 2008 – 20 p.
126. Sakabe H., Sakaguchi K., The UEC engine program and its latest development. CIMAC Congress. Kyoto. Paper No.: 224, Mitsubishi Heavy Industries, Ltd., Japan. 2004 – 8 p.
127. Bedford F., Rutland C. Effects of Direct Water Injection on DI Diesel Engine Combustion. Society of Automotive Engineers, Inc. 2000. – 10 p.

For preliminary review

SECTION 6

Marine engine oil systems

6.1 Purpose and composition of the system

The engine oil system serves for:
- supplying oil to rubbing surfaces to reduce friction between them;
- removal of heat released as a result of friction;
- removal of wear products, soot and other foreign particles from the friction zone.

In addition, the oil circulating in the system is used as a coolant to remove heat from heated parts, for example from piston heads, and also as a working fluid in hydraulic drive systems, for example gas distribution valves, hydraulic thermal gap compensators, hydraulic drive of high-pressure fuel pumps, pump-injectors, etc.

Lubricant is supplied to cylinder liners, bearings of the crankshaft and camshafts, turbochargers, pumps, drive and intermediate gears, valve guides, pushers of fuel pumps and timing mechanisms, valve rods and rocker arms. The general diagram of the oil system of a medium-speed four-stroke engine is shown in Fig. 6.1.

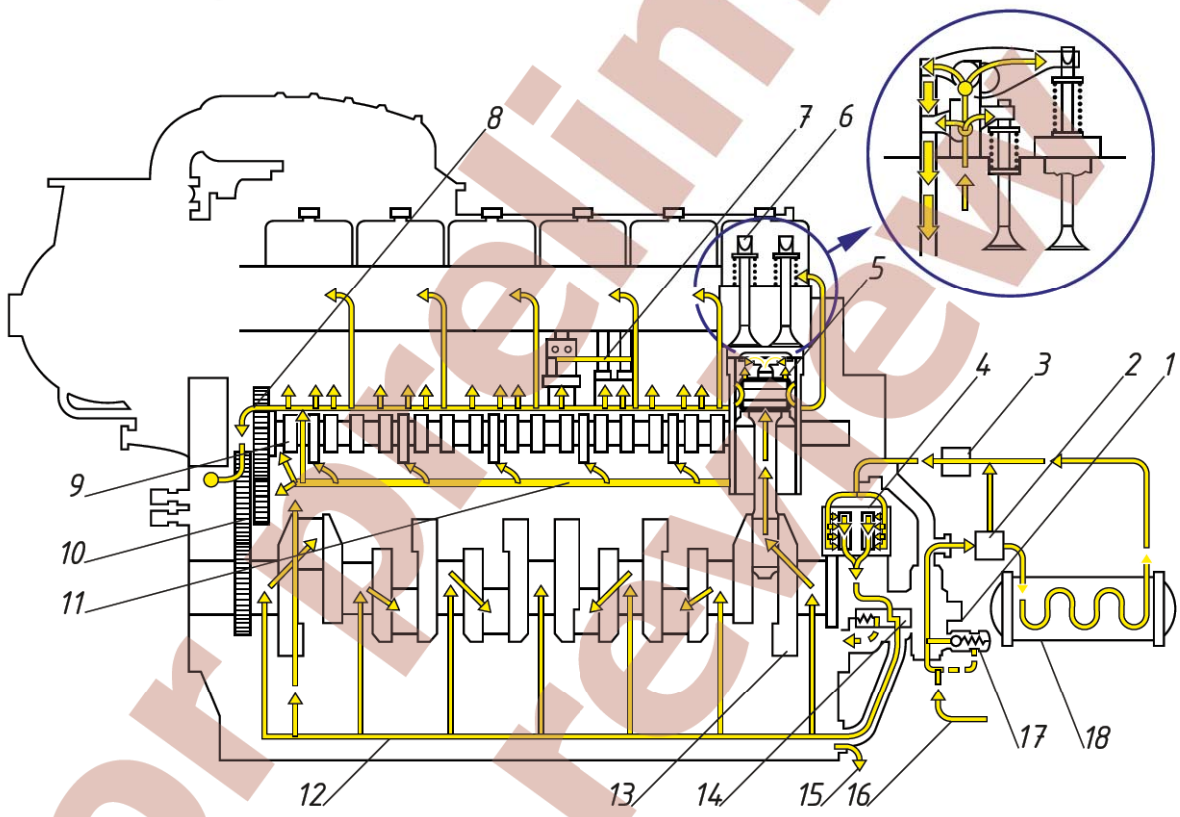


Figure 6.1 – Oil system of a four-stroke medium-speed engine TBD 645 from DEUTZ MWM: 1 – lubricating oil supply pump; 2 – lube oil thermostat; 3 – automatic filter (system side); 4 – backwash filter (engine side); 5 – pistons; 6 – valve stems; 7 – pushers valve and fuel pumps; 8 – line for supplying lubricant to valve pushers and fuel pumps; 9 – camshaft; 10 – intermediate gear; 11 – camshaft bearing lubrication line; 12 – distribution line in the oil pan; 13 – crankshaft; 14 – pressure control valve built into the backwash filter; 15 – draining lubricating oil into the tank; 16 – supply from the lubricating oil reservoir; 17 – safety valve; 18 – lubricating oil cooler (adapted from [1])

As follows from Fig. 6.1, the lubrication system includes oil tanks, oil lines, oil pumps, temperature and pressure regulators, oil filters, oil coolers (heaters).

Oil tanks are used for taking on board and storing oil reserves. Oil pumps are used to supply the required amount of oil to the discharge pipeline. Oil filters are used to clean oil from matter (wear

For preliminary review

products, carbon deposits, oil aging products). In coolers, heat is removed from the oil, which it perceives from the heated surfaces of engine structural elements.

According to the method of organizing the supply of oil to rubbing surfaces, oil systemsof marine engines are divided into open and closed (circulation).

Open-loop lubrication systemsare used in all low-speed two-stroke engines and some four-stroke engines to lubricate the friction surfaces of working cylinders. With this lubrication method, a dosed portion of oil is periodically supplied to the working surface, which is partially burned, and partially dumped by the piston rings into the sub-piston cavity, from where it is removed into the used oil tank.

In closed systems, the same oil is repeatedly supplied to friction pairs under pressure and splashing. Oil pressure is created by special pumps that are driven by the engine crankshaft or have an independent drive, most often electrical. Powerful two- and four-stroke engines are equipped with both types of pumps; a pump with a mechanical drive is used as the main one, and with an independent drive – to rock the system before starting the engine or after stopping it, as well as as a backup in case of failure of the main pump.

After the oil has worked on the friction surfaces, it flows into the oil pan.

6.2 Circulating oil systems

In small and medium power engines, all the oil used for lubrication is usually stored directly in the crankcase. This system is called a wet sump system. Its use greatly simplifies the design of the engine and the dimensions of the installation as a whole. To control the oil level in the system, the crankcase is equipped with a dipstick with appropriate graduation (Fig. 6.2).

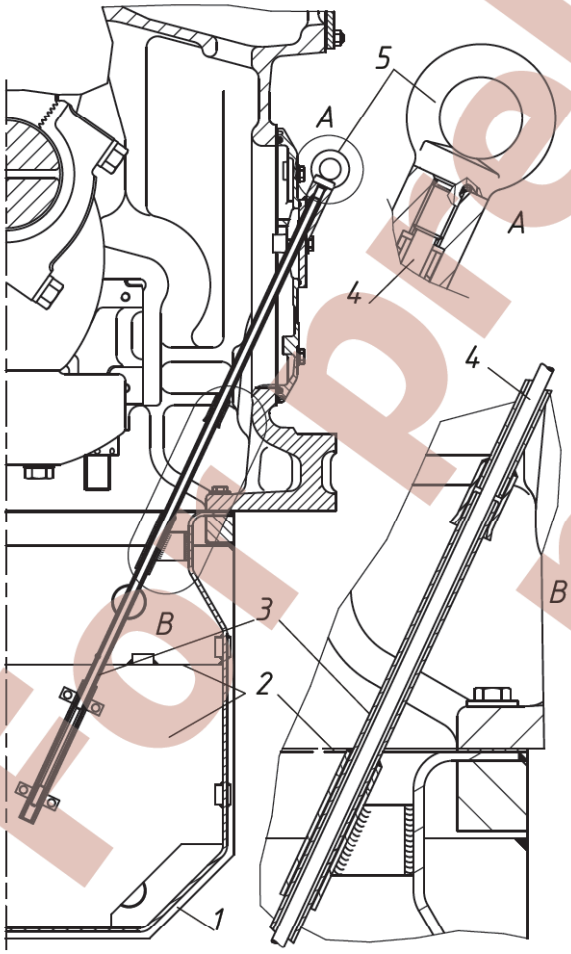


Figure 6.2 – Oil pan of the MaK M20 high-speed engine: 1 – oil pan; 2 – oil damping plates; 3 – guide tube; 4 – oil dipstick; 5 – handle for removing the probe (adapted from [2])

In order to prevent fluctuations of the oil in the crankcase during rocking of the vessel, which can lead to exposure of the oil receiver, the entire space of the pan is divided into communicating cavities using vertical and horizontal plates – oil dampers.

For preliminary review

The possibility of exposing the oil receiver when rocking the engine and the high probability of oil foaming as a result of it getting on the moving parts of the crank mechanism are the main disadvantages of dry sump systems. In addition, such engines have a greater distance from the bottom point to the axis of the crankshaft, which in some cases complicates their arrangement with other elements of the vessel's propulsion complex. In a wet sump, the oil is constantly in contact with the products of incomplete combustion that enter the crankcase through leaks in the piston rings, which leads to more intensive aging of the oil (changes in its chemical composition and physical properties).

Systems with a «dry sump» do not have the disadvantages described above, in which the oil spent in friction pairs is pumped out of the sump by a special pump (or drained) into a separate tank, where its main supply is stored. It is precisely such systemsthat have become predominantly widespread on high-power marine engines.

An important indicator of circulating oil systemsis the circulation rate, that is, the ratio of the hourly supply of circulation pumps to the volume of oil in the system. This indicator significantly affects the service life of the oil and the frequency of its replacement. Increasing the circulation rate by half reduces the oil service life by 2...4 times. For high-speed engines with a dry sump this figure is 40...60, and for high-speed engines with a wet sump it is 50...100. In medium-speed high-power engines with a dry sump, the circulation ratio is 12...30. The use of large-volume circulating oil storage tanks in dry-sump circulation systemsallows low-speed engines to provide a circulation ratio of 4…15. For this reason, the service life of the oil in low-speed engines, with regular cleaning, can reach tens of thousands of hours, and is sometimes comparable to the service life of the engine itself.

From the reserve storage tank, the circulating oil is sucked in by an oil pump through a special oil intake pipe, equipped at the inlet with a coarse filter in the form of a large metal mesh. The purpose of the filter is to prevent large objects from entering the oil pump. In some cases, to perform a similar function, a separate coarse oil filter is installed in front of the pump inlet.

Screw pumps with three working propellers are used as an oil pump on large marine engines (Fig. 6.3 *a*). On smaller engines, gear-type pumps with two (Fig. 6.3 *b*) or three working gears are predominantly used.

The performance of the pumps is selected in such a way that the oil supply covers all needs, both for lubrication of friction pairs and for cooling heated parts (for example, cooling pistons in low-speed engines), as well as for hydraulic drive of mechanisms using circulating oil as a working fluid (for example, reverse mechanism for the pump block of the RTA series engines from Sulzer (Fig. 5.38)). In this case, the oil consumption for cooling and hydraulic drive can be several times higher than the consumption associated with lubrication of engine friction pairs. Considering that during operation, oil consumption may increase, pump performance is taken with a significant margin.

Pumps in large diesel engines are usually driven by an electric motor, which allows maintaining stable operating parameters of the system regardless of the engine operating mode. In addition, this allows you to pump oil through the system when the engine is not running, for example before starting it, or for some time after stopping it to stabilize the temperatures of the cooled parts. For small engines, the oil pump is usually driven directly from the engine crankshaft through a gear transmission (Fig. 6.3 *b*). In this case, an additional bleed pump is provided, which serves to supply oil to the system when the engine is not running.

To limit system pressure, each pump is equipped with a built-in safety valve and/or a combination pressure control valve. When the pressure in the system exceeds, the oil, overcoming the force of the spring, opens the safety valve and again enters the pump inlet, which helps protect both the pump and other elements of the system from excessive loads. High-capacity pumps are equipped with pressure regulators, an example of which is shown in Fig. 6.3 *c*.

The pressure regulator uses a two-stage regulation system. An element of the first stage is a bypass valve that separates the cavity of the oil system from the inlet to the pump. A throttle insert is installed in the valve housing, through which the oil enters the closed space of the spring chamber

For preliminary review

located behind the bypass valve drive piston. Oil pressure acts on the piston and, together with the loading spring, keeps the valve closed. The pressure in the system also affects the control valve of the second stage. When the system pressure reaches the set value, the control valve opens and relieves pressure in the spring chamber of the bypass valve actuator piston. A pressure drop that does not have time to be compensated through the throttle insert leads to the opening of the bypass valve and the bypass of part of the oil from the system to the inlet of the oil pump, thereby reducing the supply pressure. If necessary, the response pressure of the regulator can be adjusted using an adjusting screw that changes the spring tension of the second stage control valve.

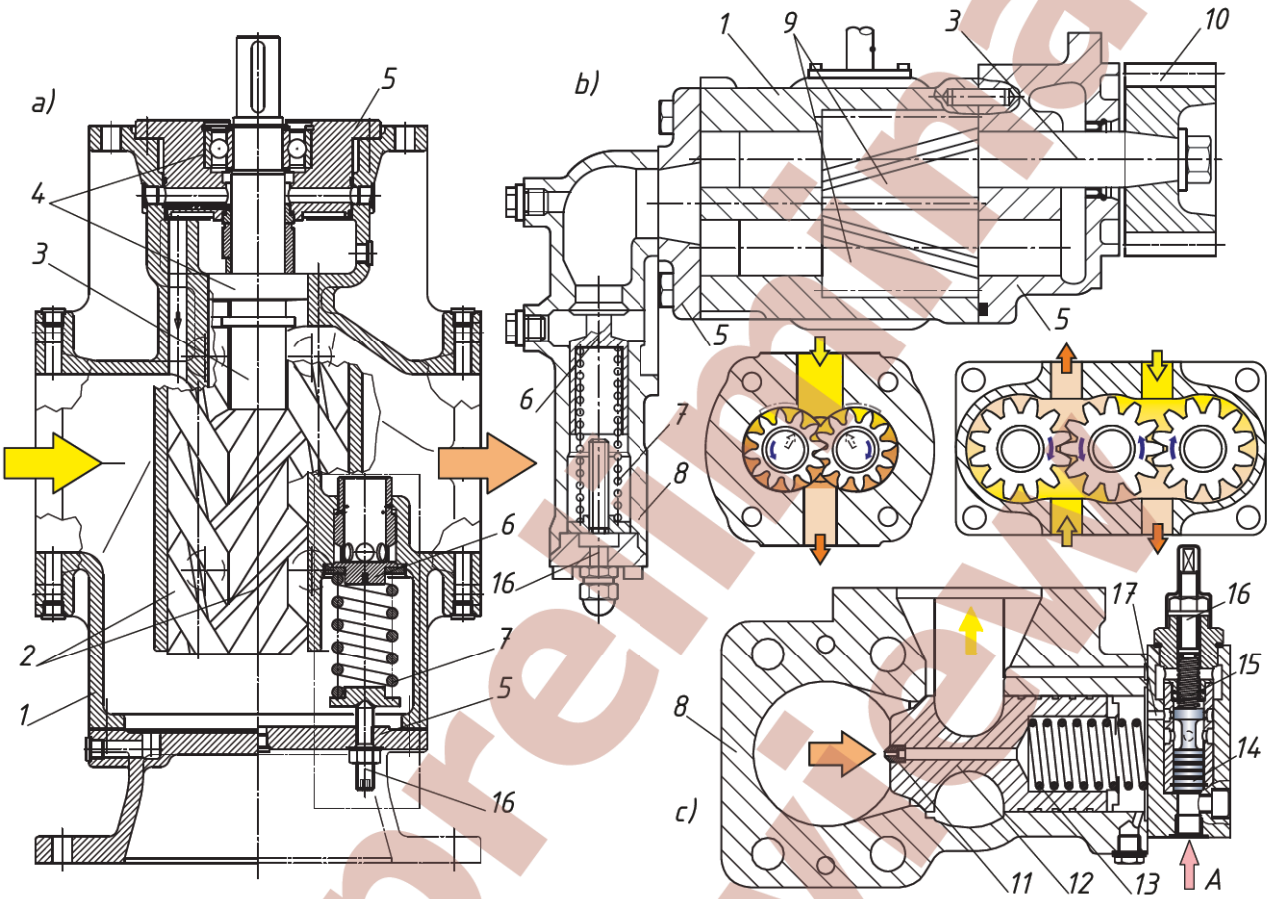


Figure 6.3 – Oil pumps of screw (*a*) and gear (*b*) types. Pressure regulator in the oil system (*c*): 1 – pump housing; 2 – pressure screws; 3 – drive shaft of pressure screws; 4 – bearings; 5 – cover; 6 – pressure reducing valve; 7 – loading spring of the pressure reducing valve; 8 – pressure reducing valve housing; 9 – pressure gears; 10 – drive gear; 11 – throttle; 12 – bypass valve; 13 – loading spring of the bypass valve; 14 – control valve; 15 – loading spring of the control valve; 16 – adjusting screw; 17 – bypass hole; A – control oil supply (adapted from [1, 3, 4])

The oil pump supplies oil to the lube oil cooler, which is equipped with a thermostat valve to maintain the set temperature in the system.

For oil systemsof powerful engines, shell-and-tube heat exchangers are used as coolers (Fig. 6.4 *a*) in which cooled oil moves through pipes (pipe space), and cooling water in the inter-tube space. In this case, the heat from the hotter oil is transferred to the cooling water through the surface of the pipe walls. To allow expansion in the longitudinal direction, the pipe package is inserted into the jacket and secured at one end. Most often, the opposite direction of movement of coolants is provided, which promotes the most efficient heat exchange. The heat exchanger tubes are located in the core of the casing and have a direct impact on the speed of movement of the substance. Special partitions are installed in the casing that change the direction of water movement, which helps to increase the heat transfer coefficient and the efficiency of the heat exchanger (Fig. 6.4 *b*). The oil cooler pipe package is made of copper-nickel alloy, and the water chambers are cast from cast iron or welded from sheet steel.

For preliminary review

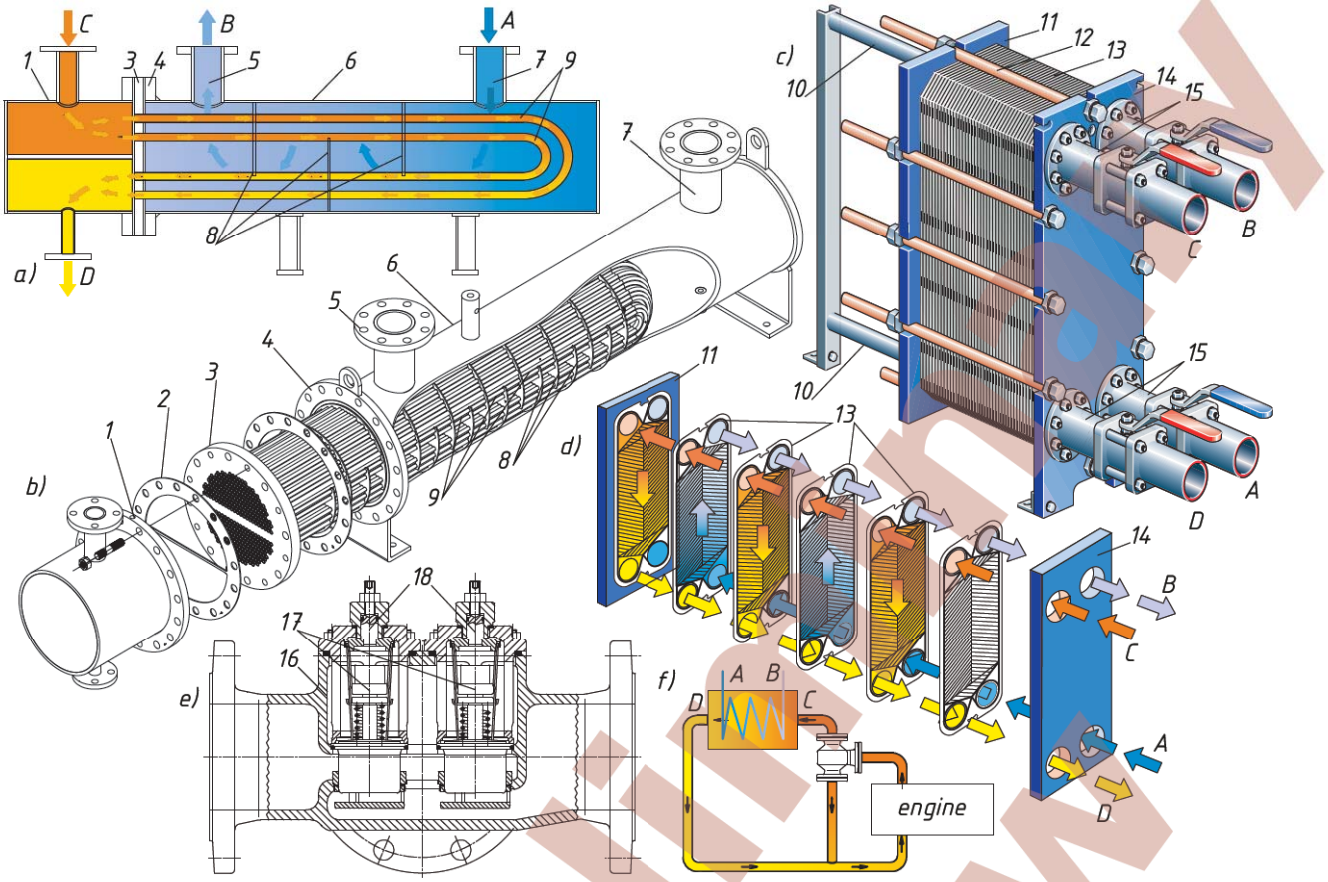


Figure 6.4 – Heat exchangers and oil temperature regulators: *a* – diagram of the movement of fluid flows in a shell-and-tube oil heat exchanger with U-shaped tubes; *b* – general design of a shell-and-tube oil heat exchanger; *c* – general view of the plate oil heat exchanger; *d* – diagram of the movement of liquid flows in a plate heat exchanger; *e* – oil temperature regulator;*f* – regulator switching circuit; *A* – cooling water inlet; *B* – cooling water outlet; *C* – hot oil inlet; *D* – cooled oil inlet; 1 – cover with dividing partition; 2 – gasket; 3 – tube sheet; 4 – flange; 5 – hot oil supply pipe; 6 – housing; 7 – cooled oil outlet pipe; 8 – guide partitions; 9 – heat exchange tubes; 10 – supporting rod; 11 – pressure plate; 12 – guide rods; 13 – set of plates; 14 – front plate; 15 – connecting pipes; 16 – thermostat housing; 17 – thermostats; 18 – adjusting screws (adapted from [4, 5, 6])

For engines of medium and low power, plate heat exchangers are used (Fig. 6.4 *c*), the operation of which is based on heat exchange between oil and water through a contact plate. The coolants move through the internal space of the heat exchanger, formed by a package of plates (Fig. 6.4 *d*) which form two movement circuits due to the alternation of plates with dense rubber gaskets. The heat exchanger plates are made of stainless steel or titanium. The number of plates depends on the technical parameters and requirements for the productivity of the heat exchanger.

The plate pack is sandwiched between two pressure plates and tightened using tie rods. For ease of assembly, disassembly and maintenance of the device, supporting rods are used, on which the plates are placed, and supports, to which the remaining structural elements are attached.

Due to the high speed of the working media inside plate heat exchangers, deposits accumulate on its internal surfaces more slowly than on the surfaces of shell-and-tube devices. The possibility of complete disassembly allows not only washing the plates, but also their mechanical cleaning, and, if necessary, partial or complete replacement of damaged elements.

On low-power engines, plate heat exchangers of a non-separable design are used, in which a package of plates is connected by soldering. Such heat exchangers are cheaper than collapsible ones, but their maintenance and repair are difficult.

Regardless of the type of heat exchanger used in the system, the pressure in the oil circuit is always maintained 0.01...0.05 MPa higher than in the water circuit. This allows, in the event of a heat

For preliminary review

exchanger depressurization, to avoid cooling water from entering the lubricating oil, which can lead to serious engine damage.

Fresh water is most often used as a cooling medium for heat exchangers, which circulates in a closed loop and transfers the removed heat to sea water in the external loop heat exchangers.

To regulate the oil temperature, a thermostatic valve is installed in the system (Fig. 6.4 *e*). The valve housing contains two thermostats adjusted to the set opening temperature. When the temperature of the incoming oil exceeds the nominal value, the stearic filler located inside the thermostats begins to melt, significantly increasing in volume. As a result of this, a rod is forced out of the thermostat housing, driving the valve block, which reduces the flow area of the channel, redirecting part of the oil to the cooler (Fig. 6.4 *f*). This movement continues until the desired temperature of the mixed oil leaving the engine is reached.

Regulating the oil temperature allows you to increase the efficiency of its cleaning in the oil filter, through which the oil passes before entering the engine. There are a large number of oil filter designs, but on marine engines the most common are full-flow filters, through which the entire oil flow passes before being supplied to the engine.

The most effective filtration (cleansing oil from foreign impurities) is achieved by paper filter elements, in which the oil passes through the porous structure of special paper, which traps all foreign particles.

The filter elements are placed in housings that have special channels for supplying unrefined oil to the filter elements and removing purified oil from them. Several filter elements can be placed in one housing. In marine engines, so-called duplex filters are widely used (Fig. 6.5 *a*), consisting of two filter chambers (in some designs there may be more filter chambers). The duplex filter housing is equipped with a three-way valve that allows you to switch oil flows through separate chambers. This allows you to replace filter elements in one of the chambers without stopping the engine by switching the oil flow to another filter chamber.

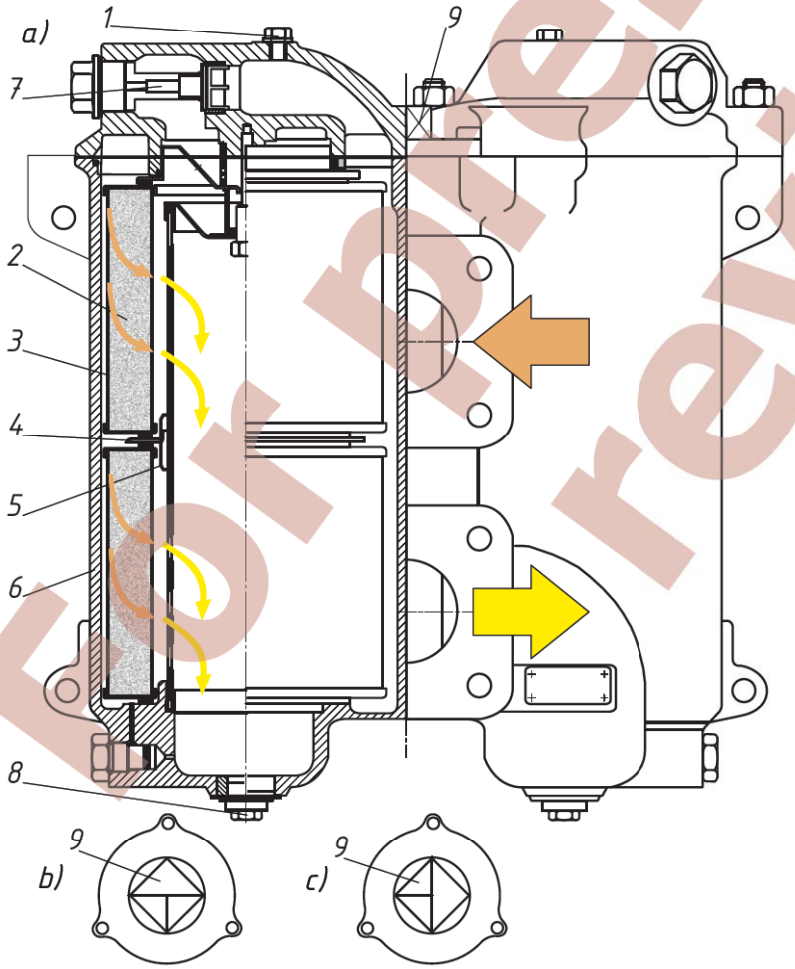


Figure 6.5 – Duplex oil filter for 32 series engines from Wartsila (*a*): 1 – air release plug; 2 – filter element; 3 – metal mesh; 4 – spacer; 5 – metal sleeve; 6 – housing; 7 – bypass valve; 8 – plug for draining sediment; 9 – three-way valve; *b* – normal position of the three-way valve, both sections of the filter are in operation; *c* – the right section of the filter is turned off for maintenance; *d* – general view of the filter element (adapted from [7])

For preliminary review

In normal mode, oil passes through both filter chambers, which corresponds to the position of the mark on the spindle of the three-way valve as shown in Fig. 6.5 *b*. When replacing the filter element, the spindle is moved to the position shown in Fig. 6.5 *c*. In this case, the right filter chamber is taken out of operation, the cover can be removed from it and the filter elements can be replaced.

Filter elements (Fig. 6.5 *d*) are made of special paper that is capable of trapping particles larger than 15 μm. The filter element frame is made of metal mesh and stamped covers, which are installed on a perforated sleeve. These elements provide additional protection in case the paper liner fails or bypasses. The housings of the filter chambers are equipped with bypass valves, which open when the pressure drop across the filter element exceeds 0.2...0.3 MPa, which indicates its clogging. In this case, unrefined oil enters the engine. Modern engine filters are equipped with visual indicators and/or electrical switches connected to an automatic alarm system to warn of high filter pressure drop.

In recent years, automatic self-cleaning filters have begun to be used on marine engines as a stand-alone unit or in addition to the traditional filter elements discussed above. An example of such a unit produced by Boll Filters is shown in Fig. 6.6.

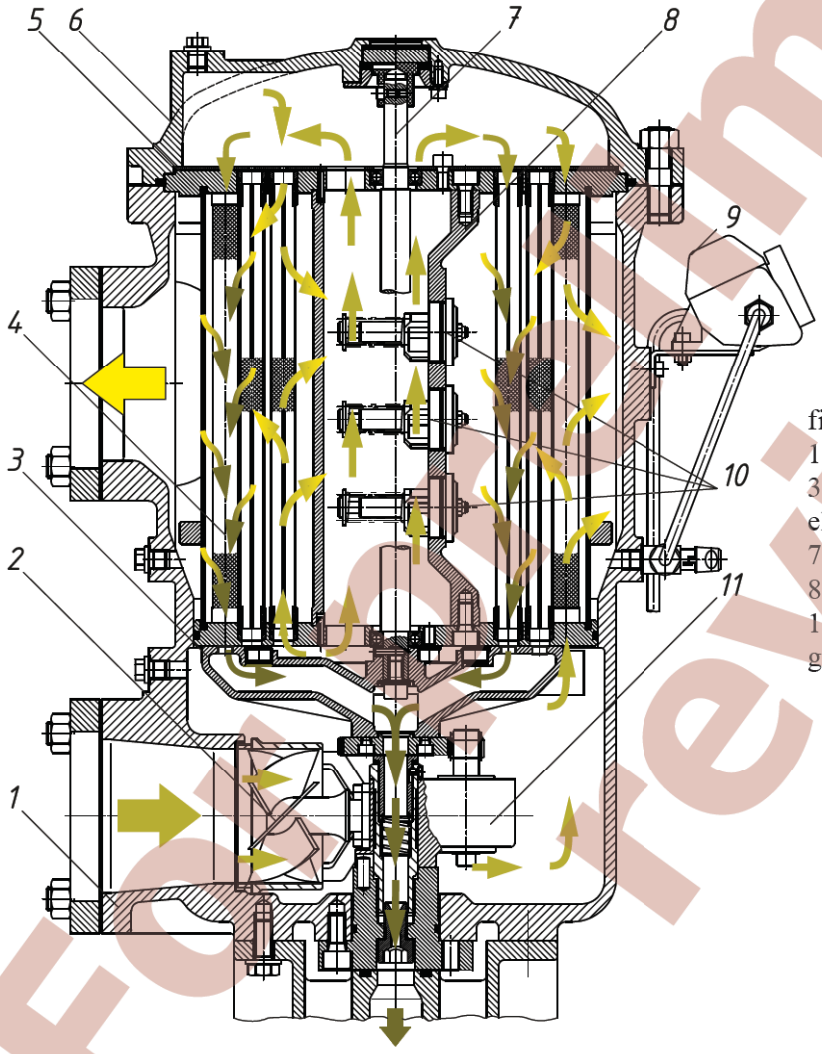


Figure 6.6 – Automatic self-cleaning filter series 6.46 from Boll Filters: 1 – filter housing; 2 – drive impeller; 3 – sludge collection support; 4 – filter elements; 5 – top board; 6 – filter cover; 7 – axis of rotation of the caliper; 8 – diaphragm; 9 – control panel; 10 – bypass valves; 11 – caliper drive gearbox (adapted from [8])

The filter consists of a housing in which two metal boards with holes for installing slot-type filter elements are clamped at the top and bottom. Each element is a metal sleeve made of perforated metal, on the surface of which a metal mesh with a special weave is wound, capable of retaining particles larger than 25 μm. Due to their elongated cylindrical shape, these filter elements are called filter candles.

During operation of the filter, oil passes through the inlet flange, in which a hydraulic turbine wheel is installed to drive the flushing device. After the turbine, part of the unrefined oil flow enters

For preliminary review

the filter candles through holes in the bottom plate. The second part of the flow through the cen-tral riser is supplied to the upper part of the filter housing and then enters the filter cartridges through holes in the upper board. Thus, the oil enters the filter cartridges on both sides, which en-sures an even distribution of contaminants along the height of the filter candles.

After passing through the filter candles, the purified oil enters the central part of the housing, forming a collection manifold, from where it flows either directly into the engine or for additional cleaning.

For automatic, continuous cleaning of the filter from sludge, a rotating caliper is installed in the lower part of the housing, driven through a gearbox from a hydraulic turbine wheel. The caliper is pressed against the end surface of the lower board and performs the function of a spool. Its housing has channels that, when rotated, periodically disconnect the filter elements from the oil supply cav-ity. As a result of this, the pressure in the filter element drops sharply, as a result of which the oil flow changes direction, washing the cracks of the filter material in a countercurrent manner. Sludge en-ters the caliper cavity and is then discharged out through its axis. Uniform backflushing over a long period of time is achieved by using a small amount of flushing oil and prevents a drop in oper-ating pressure in the system.

In case of filter clogging, the housing is equipped with a separating diaphragm with pressure re-ducing valves installed on it. When the pressure difference increases to 0.2 MPa between the supply cavity and the collection manifold, the valves open and allow unrefined oil to flow into the system. In this case, on the control panel attached to the filter housing, the control zone is painted red. At the customer's request, the control panel can be equipped with light and sound alarms.

As an addition to the main filter, some four-stroke engines are equipped with a centrifugal-type filter, the so-called centrifuge. The most widespread on marine engines are bypass-type centrifuges that purify circulating oil without supplying it directly to the engine oil system (Fig. 6.7). However, there are designs in which centrifuges are used to purify the flushing oil after the automatic filter.

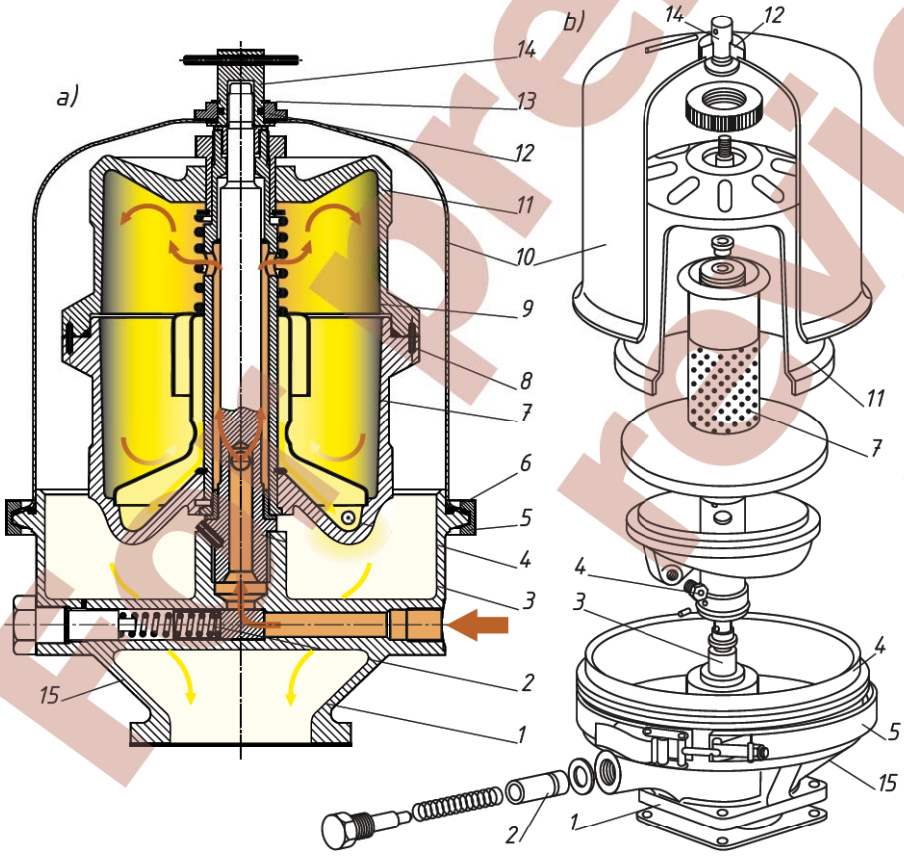


Figure 6.7 – Centrifugal oil purifiers (centrifuges): *a* – gen-eral design; *b* – main elements; 1 – purified oil drainage cavity; 2 – safety valve; 3 – rotor axis; 4 – nozzle; 5 – coupling clamp; 6 – seal; 7 – rotor hub; 8 – guide pin; 9 – spring; 10 – cover; 11 – rotor housing; 12 – bush-ing; 13 – retaining ring; 14 – cover fastening nut; 15 – housing (adapted from [9])

A characteristic feature of the design is that the rotor rotates on a vertical fixed axis in plain bearings.

Preliminary review

The filter consists of a housing in which a hollow axis is installed, on which a dynamically balanced rotor mounted on plain bearings rotates freely. Part of the oil supplied by the pump (approximately 10% of the total volume) flows through the housing along a hollow axis into the rotor. The rotor consists of two cavities: the cleaning cavity and the drive cavity. From the axle, the oil enters the upper part of the rotor, where it is exposed to high centrifugal force. Dirt is deposited on the ro-tor walls in the form of heavy sediment. The oil then flows from the cleaning cavity into the drive cavity formed by the riser and the lower part of the rotor. To drive the rotor, a reactive force is used, created by two tangentially located nozzles at the bottom of the drive cavity. As the purified oil passes through the nozzles on the rotor, a torque is created, after which the oil is returned through the filter housing to the engine crankcase.

In addition to the filters already described, magnetic filters can be installed in the oil system to capture small metal particles. They can be made either as separate units or integrated into existing design elements of oil system units.

For powerful two-stroke crosshead engines, the circulating oil is cleaned using special centrifugal cleaners – separators. These devices are independent units, so their consideration is beyond the scope of this book. To familiarize yourself with these devices, we advise you to consult specialized literature.

The purified oil enters the engine circulation system, which consists of oil lines laid in the engine crankcase and serving to supply oil to the most loaded friction surfaces. In powerful two-stroke engines, the oil lines are made in the form of separately laid pipelines. In four-stroke engines, most of the oil lines are drilled in the crankcase.

In four-stroke engines, lubricating oil is supplied to the main bearings, from which it is supplied to lubricate the connecting rod bearings through special drillings in the journals and cheeks of the crankshaft (Fig. 6.8). There are oil pockets on the bottom and sides of the bearing shells (less loaded parts). From these, oil is supplied through a drilling in the connecting rod to lubricate the upper bearing of the connecting rod, to lubricate the piston group and to cool the pistons.

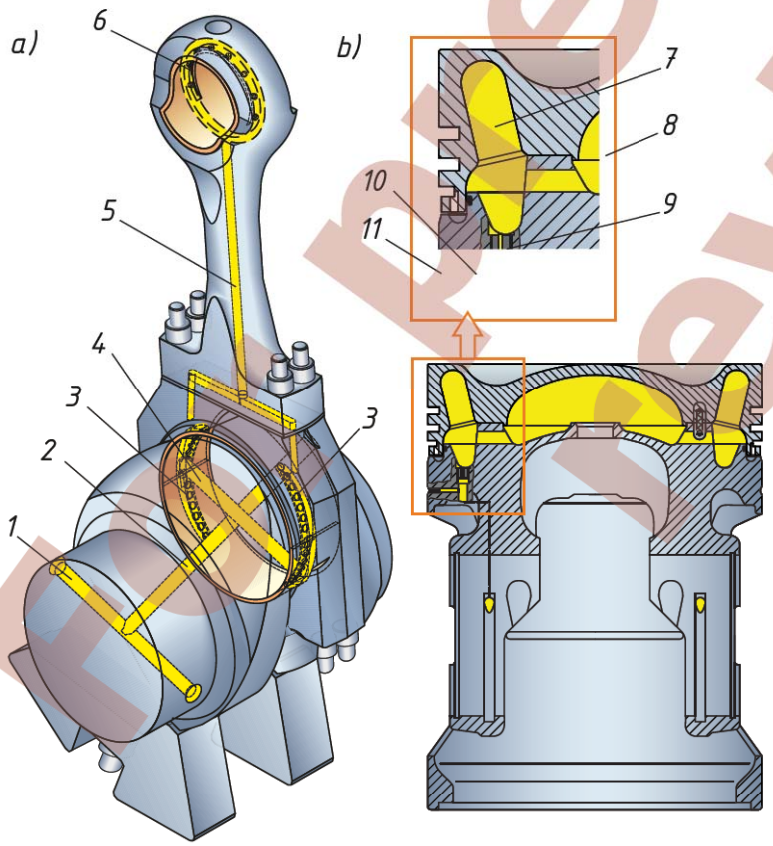


Figure 6.8 – Lubrication of bearings of the crank mechanism and the cylinder-piston group of a four-stroke engine: 1 – radial channel for collecting oil from the oil pocket of the main bearing; 2 – inclined channel for supplying oil to the main bearing; 3 – oil pockets of the main bearing; 4 – radial channel for supplying oil to the main bearing pockets; 5 – channel for supplying oil to the upper bearing of the connecting rod; 6 – oil pocket of the upper bearing of the connecting rod; 7 – cooling cavity of the piston head in the area of the piston rings; 8 – cool-ing cavity of the fire bottom of the piston; 9 – throttle insert; 10 – channel for supplying oil to the nozzles insert; 11 – nozzles insert for supplying oil to the cylinder liner (adapted from [4, 10])

In addition to the lubrication points already mentioned, the following are also lubricated under pressure: injection pump pushers and gas distribution valves; camshaft bearings; push bars; rocker

For preliminary review

arm axes for timing valve drives, etc. The remaining elements are lubricated by splashing (for example, camshaft cams) or by jet (for example, drive gears).

The circulating oil system of low-speed two-stroke engines usually consists of two separate circuits: low and medium pressure. The main points for supplying circulating oil to the friction surfaces of two-stroke crosshead engines are shown in Fig. 6.9.

The low-pressure circuit provides lubrication of frame and thrust bearings, drive gears, camshaft bearings, fuel pumps, chain (gear) drive, as well as cooling of the crosshead parallels. The medium pressure system lubricates the head and crank bearings, crosshead sliders, and also supplies oil for cooling the pistons. Low and medium pressure systemsare served by one oil pump, creating a pressure of 0.4...0.8 MPa. Pressure regulators are used to regulate the pressure in each circuit. Some designs provide for the installation of an additional pump between the low and high pressure circuits, which increases the pressure from low (0.28...0.36 MPa) to medium (0.8...1.2 MPa).

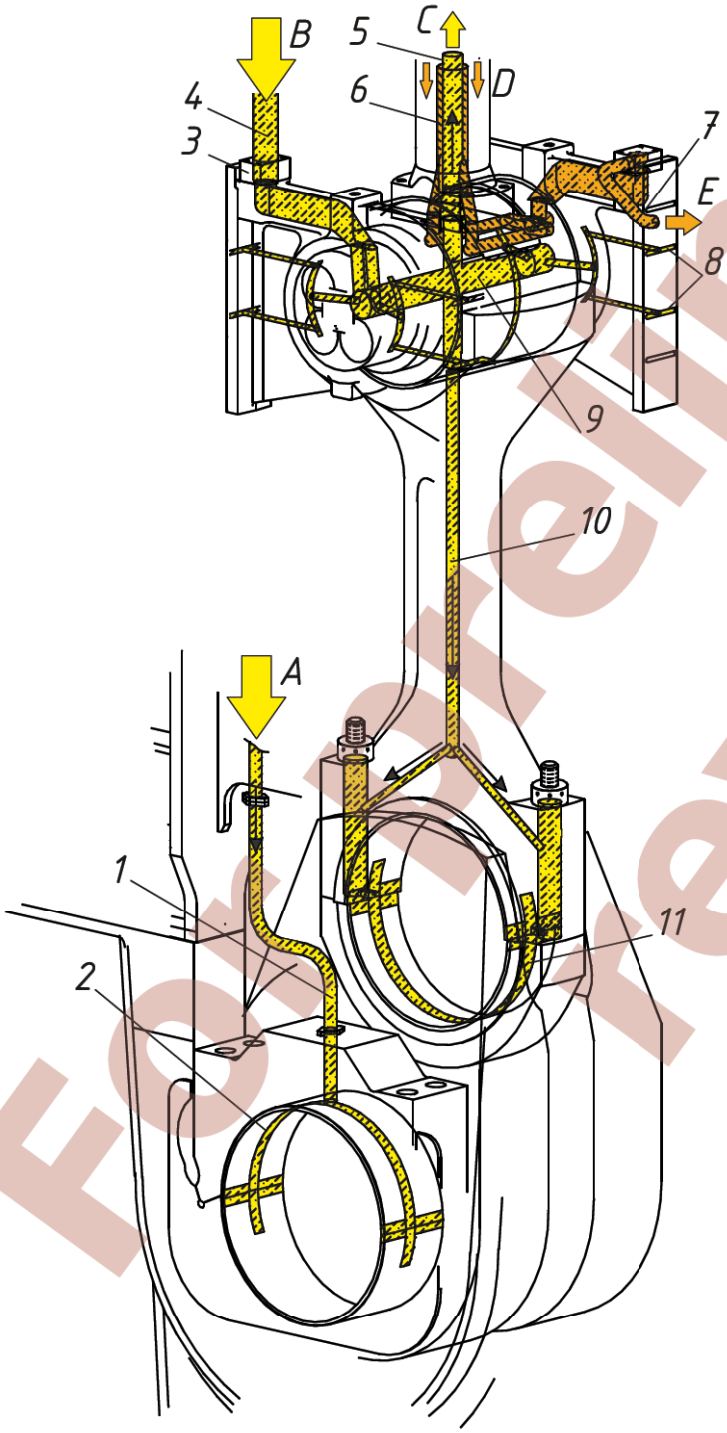


Figure 6.9 – Oil supply to the friction pairs of the crank mechanism and for cooling the pistons: *A* – oil supply to the frame bearings; *B* – oil supply to the crosshead bearings and sliders; *C* – oil supply for cooling the pistons; *D* – removal of heated oil from the piston; *E* – draining the cooling oil into the engine crankcase; 1 – channel for supplying oil to frame bearings; 2 – oil distribution groove of the frame bearing; 3 – flange for fastening the telescopic oil supply rod; 4 – telescopic hose; 5 – oil drainage channel for piston cooling; 6 – channel for draining cooling oil from the piston; 7 – cooling oil drain pipe; 8 – oil supply to the slider shoes and parallels; 9 – oil distribution manifold of the crosshead unit; 10 – channel for supplying oil to the lower bearing of the connecting rod; 11 – oil distribution groove of the lower bearing of the connecting rod (adapted from [11])

For preliminary review

In this case, the low and medium pressure oil systemsare connected to each other through a check valve. If the medium pressure oil pump fails, the crosshead bearing oil system is supplied with oil pressure from the main bearing. Under such conditions, the engine can only be operated at reduced load.

Crankshafts of low-speed diesel engines usually do not have drillings for oil supply. Lubricant to the frame bearings is supplied from the low-pressure oil pipeline passing inside the diesel frame. Oil flows to the crank bearings through a drilling in the connecting rod from the crosshead assembly.

To supply oil to the moving crosshead assembly of low-speed engines, telescopic pipes (MAN and Mitsubishi) or toggle mechanisms (Sulzer, Wärtsilä, WinGD) are used. The general structure of the toggle mechanism, as well as the order of its operation, are shown in Fig. 6.10 *a*. The telescopic mechanism for supplying oil to the crosshead unit is shown in Fig. 6.10 *b*.

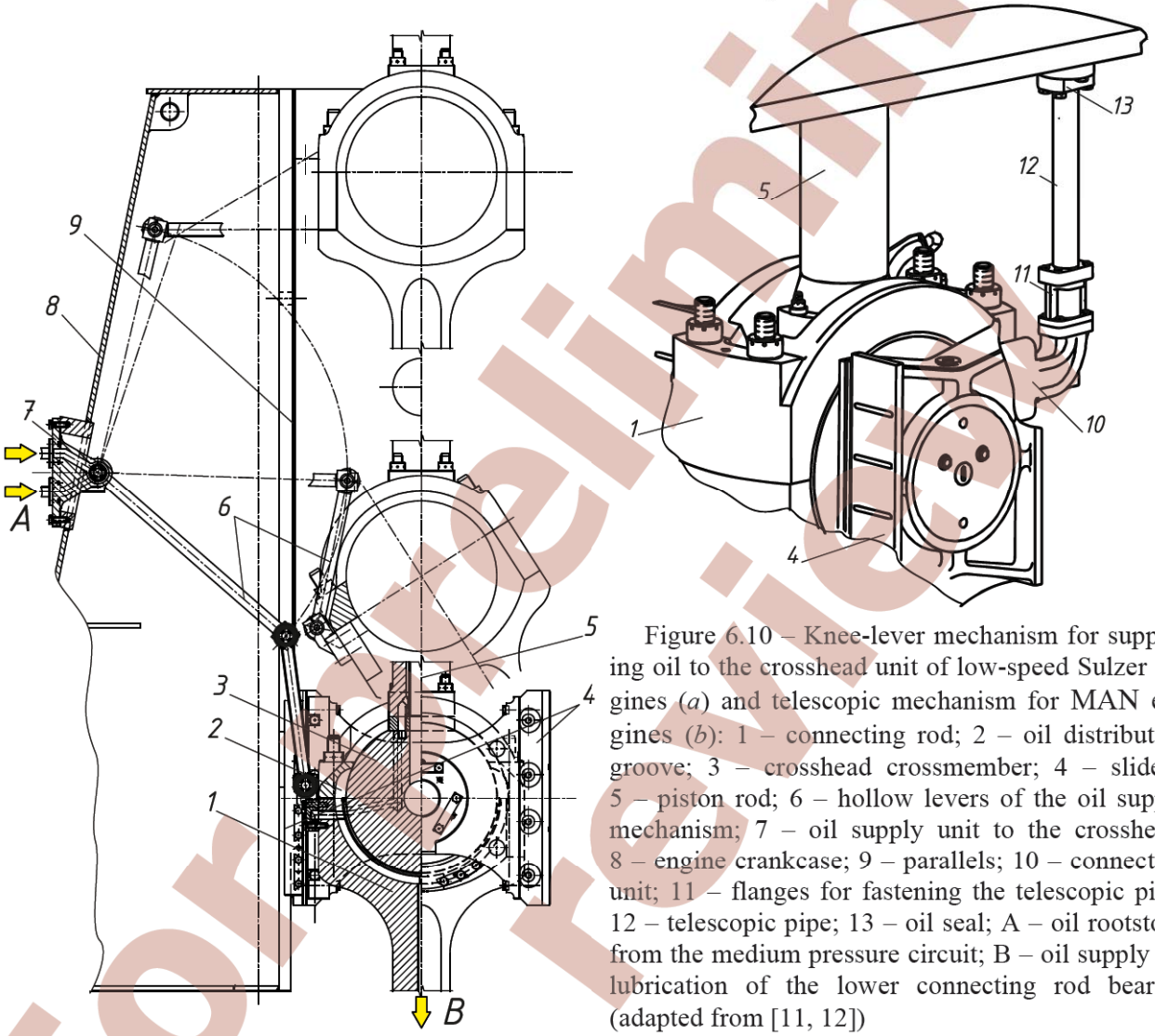


Figure 6.10 – Knee-lever mechanism for supplying oil to the crosshead unit of low-speed Sulzer engines (*a*) and telescopic mechanism for MAN engines (*b*): 1 – connecting rod; 2 – oil distribution groove; 3 – crosshead crossmember; 4 – sliders; 5 – piston rod; 6 – hollow levers of the oil supply mechanism; 7 – oil supply unit to the crosshead; 8 – engine crankcase; 9 – parallels; 10 – connecting unit; 11 – flanges for fastening the telescopic pipe; 12 – telescopic pipe; 13 – oil seal; A – oil rootstock from the medium pressure circuit; B – oil supply for lubrication of the lower connecting rod bearing (adapted from [11, 12])

For engines with timing valves driven by a camshaft, oil is supplied to their hydraulic drive system from the medium pressure oil system through an air separator. Further, at the moment the valves open, the actuators raise the oil pressure in the system to approximately 16 MPa. Other hydraulically driven engine mechanisms are typically driven by oil from a medium pressure system.

6.3 Cylinder lubrication systems

Lubricating lubrication systemsused on powerful two-stroke and some four-stroke engines are open-loop, that is, oil is supplied once to the friction surface of the working cylinder, where it is irretrievably lost (burns) or flows into the waste oil tank collection.

For preliminary review

The presence of a separate lubrication system for the working cylinders of low-speed two-stroke crosshead engines is associated with their design features:

- the specific nature of the organization of gas exchange processes leads to the need to divide the internal volume of the engine into two cavities – the crankcase and the sub-piston. This eliminates the possibility of using oil mist generated in the crankcase cavity to lubricate the working surfaces of the cylinder liners;
- the mechanisms and conditions for the occurrence of physical and chemical processes in the cylinder-piston group differ significantly from those operating in the crank mechanism, which makes it preferable to use oils specially adapted for these conditions;
- the use of heavy fuels with a high sulfur content, on the one hand, leads to the rapid accumulation of products of incomplete combustion and aging of the oils themselves, and on the other hand, to the need to neutralize sulfur combustion products, which have an increased corrosive effect.

The cost of lubricating oil for working cylinders significantly affects the cost of operating the engine. In addition, the reliability and safety of engine operation significantly depends on the efficiency of lubrication systems.

In this regard, significant efforts have been devoted to the development and implementation of new types of systemsin recent years. Their goal was to reduce lubricating oil consumption while maintaining satisfactory wear rates for piston rings and cylinder liners. In turn, reducing lubricating oil consumption also has a positive impact on the environment, as it leads to a reduction in harmful emissions.

The previously used mechanical systemsfor metered oil supply to engine-driven working cylinders have been almost completely replaced by new electrically or hydraulically driven systems, which make it possible to regulate the supply of lubricating oil over a wider range depending on engine operating conditions and other operational factors.

A fairly common approach today in the field of constructing lubrication systemsfor working cylinders is the use of mechanically driven pump modules, which have proven themselves quite well in operation (Fig. 6.11). To expand control capabilities, such blocks serving individual engine cylinders are assembled on a common platform. They are driven through a gearbox from an electric motor with frequency control.

The pump block includes several pump sections, each of which pumps lubricating oil to a separate supply valve to the working cylinder (Fig. 6.11). All sections are driven by a common camshaft.

The lubricating oil stock is located in the pump block housing, from where it is supplied to the pump element. The direction of oil flow is regulated using ball valves installed in the discharge element housing. During the suction stroke, the plunger of the discharge element increases the volume of the discharge cavity, which is filled with oil through the opened lower ball valve. The magnitude of the discharge stroke of the plunger depends on the initial position of the double-arm drive lever, which is adjusted by turning the eccentric axis of the lever and the stop in the upper part of the housing. By changing the position of the stop, you can quite accurately adjust the performance of the pumping element.

During the pumping stroke, the plunger forces oil into the glass rotameter through automatically opening ball valves. The transparent rotameter tube allows you to visually monitor the oil flow, and by the position of the pointer ball you can evaluate the performance of the pumping element and, if necessary, adjust it with the flow regulator by changing the position of the stop of the double-arm lever.

From the rotameter, the oil flows to the fitting on the pump housing, to which the oil supply tube to the supply valve to the working cylinder is attached.

Each pump element is equipped with a safety valve to protect the pump section in case the outlet on the slave cylinder oil supply valve becomes blocked. When the safety valve is activated, the pumped lubricating oil is returned to the pump's flow cavity.

The use of an electric drive with frequency control and phase synchronization of the supply relative to the position of the engine crankshaft allows oil to be supplied to the working cylinders in a

For preliminary review

clearly defined area. For example, when passing a set of piston rings or a piston skirt, relative to the oil supply valves to the cylinders.

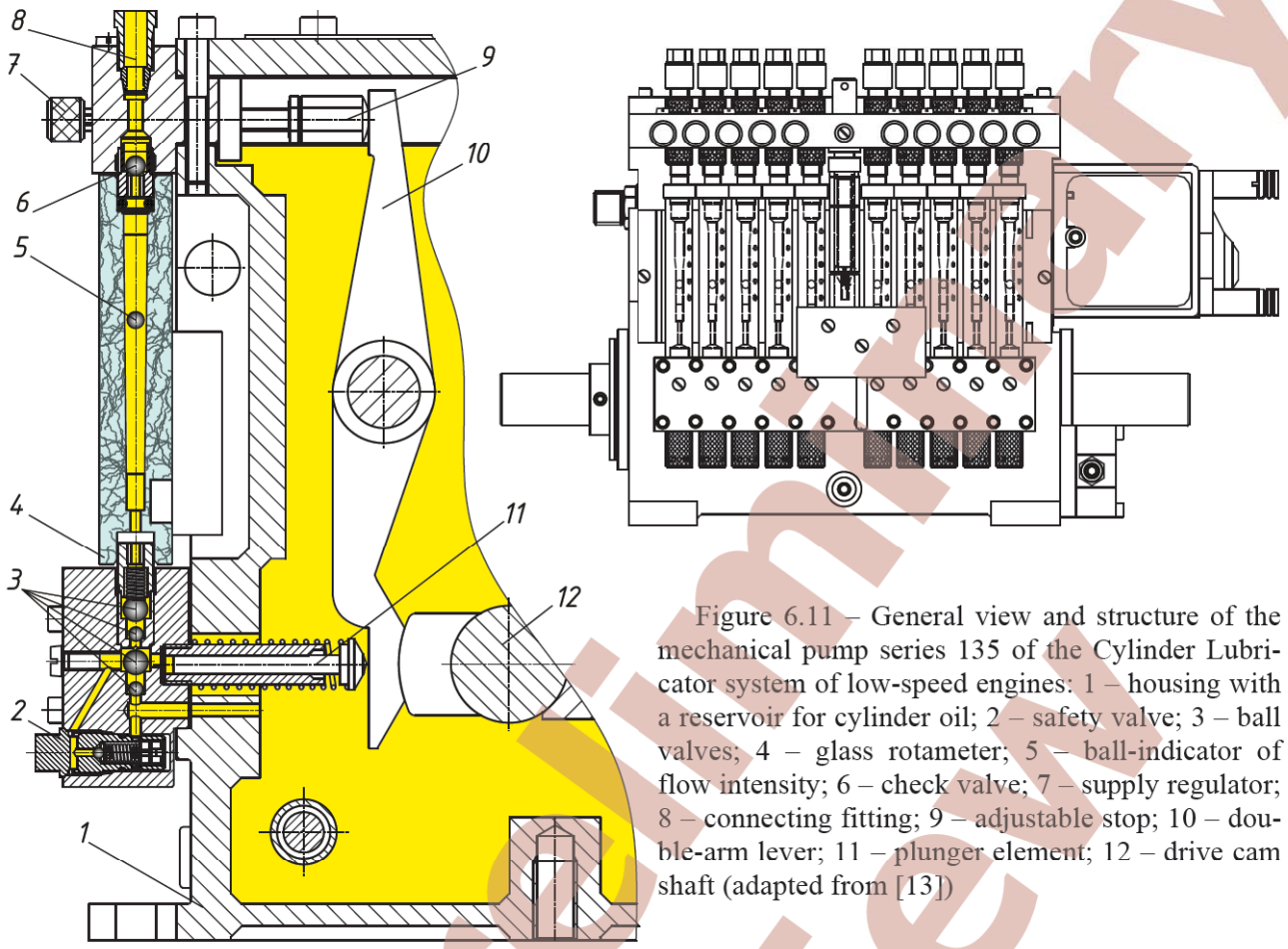


Figure 6.11 – General view and structure of the mechanical pump series 135 of the Cylinder Lubricator system of low-speed engines: 1 – housing with a reservoir for cylinder oil; 2 – safety valve; 3 – ball valves; 4 – glass rotameter; 5 – ball-indicator of flow intensity; 6 – check valve; 7 – supply regulator; 8 – connecting fitting; 9 – adjustable stop; 10 – double-arm lever; 11 – plunger element; 12 – drive cam shaft (adapted from [13])

The disadvantage of the design described above is the presence of rather long oil supply lines from the pump unit to the supply valves on the working cylinders, the number of which can reach from eight to sixteen per cylinder (on large engines, the oil supply valves are installed in two tiers).

The number of lines can be significantly reduced by using so-called progressive distributors, which are mounted directly on the cylinder block and provide a sequential supply of lubricating oil to several supply valves connected to them.

In Fig. 6.12 shows a CLU3 type cylinder lubrication system in which the principle described above is implemented. This system has become widespread on Sulzer RTA series engines. The pump block consists of pump sections, each of which serves two engine cylinders. If the engine has an odd number of cylinders, then plugs are installed in one section instead of pumping elements (Fig. 6.12 *b*). Each section is equipped with two horizontally and two vertically located pumping elements (Fig. 6.12 *a*). Vertical elements provide oil supply to the supply valves of the first tier, horizontal elements of the second tier. All pumping sections are driven by an electric motor with frequency regulation and phase synchronization with the position of the crankshaft.

The pump elements (Fig. 6.12 *e*) are equipped with bushings that, due to a threaded transmission, can change their position relative to the edge of the plunger. As a result, the moment the plunger edge closes the filling hole changes, which leads to a change in the active stroke of the plunger and the performance of the pumping element. The bushing has six fixed positions, which are set by turning the running nut and secured with a locking screw that fits into the corresponding recess on the lock washer (Fig. 6.12 *c*). Thus, the performance of the pumping unit is pre-set, depending on the type of engine on which it is installed.

Each pumping section supplies oil through a pipeline to a progressive distributor, which is a

For preliminary review

complex hydromechanical device that allows oil to be supplied to two valves on the cylinder liner in a strictly defined order each time the plunger is pumped (Fig. 6.12 *d*). During this cycle, special pistons installed in the distributor dispense 0.35 ml of lubricating oil regardless of viscosity, back pressure and speed. This process is repeated continuously (progressively).

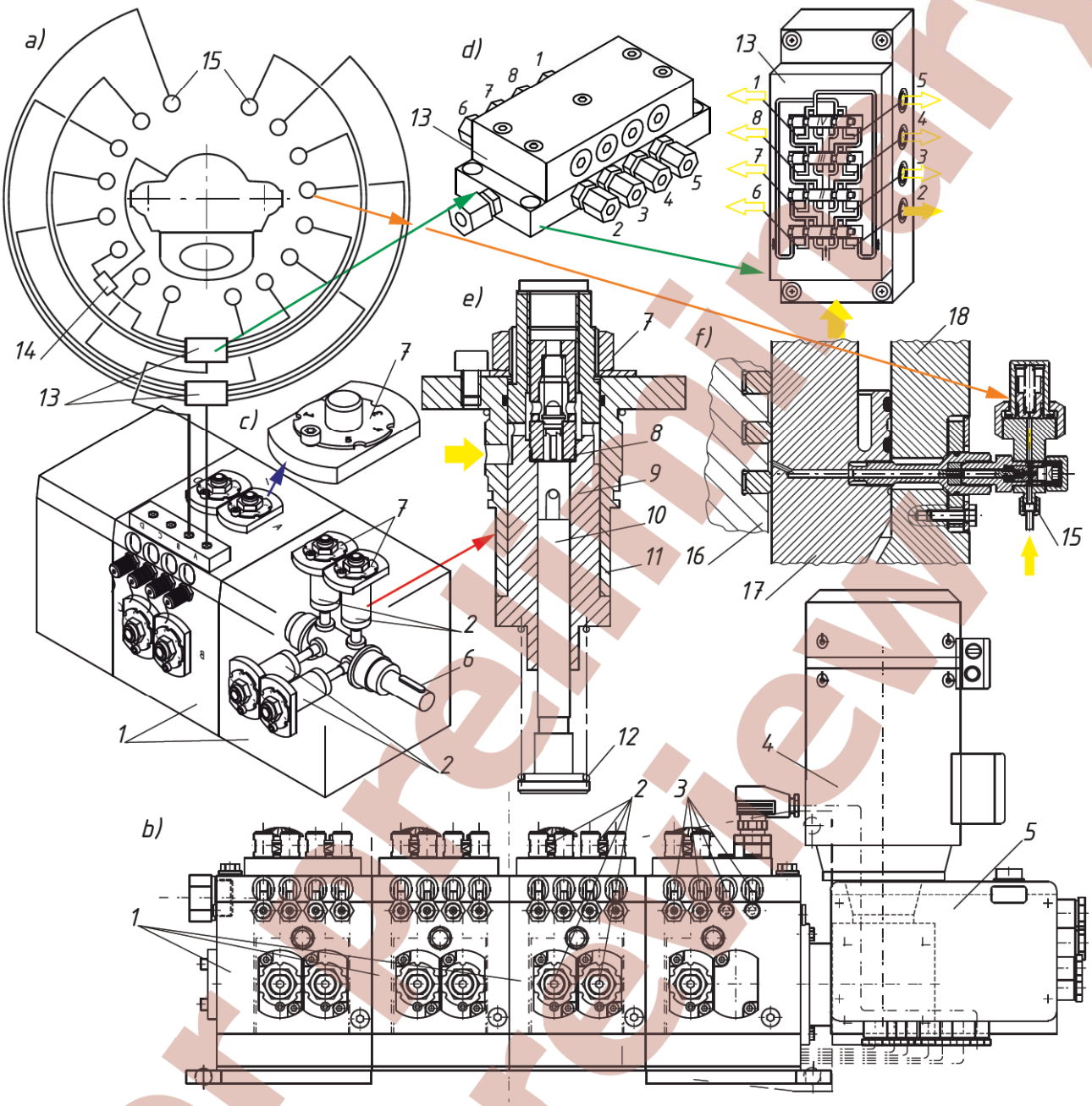


Figure 6.12 – CLU3 type cylinder lubrication system for two-stroke low-speed engines with oil supply valves and spring pressure accumulator equipment. *a* – general diagram of the system; *b* – general view of the pump unit; *c* – mechanism for adjusting the performance of the pump sections; *d* – lubricating oil distributor; *e* – plunger element; *f* – installation of the oil supply valve on the working cylinder. 1 – pumping section; 2 – plunger element; 3 – control rotameters; 4 – electric motor with gearbox; 5 – electrical switching unit; 6 – drive shaft; 7 – scale for adjusting the performance of the plunger element; 8 – check valve housing; 9 – bushing of the discharge element; 10 – plunger; 11 – plunger element housing; 12 – return spring; 13 – lubricating oil distributor; 14 – oil distributor for lubrication of the exhaust valve module; 15 – valve for supplying lubricating oil to the working cylinder; 16 – piston with a set of piston rings; 17 – working cylinder bushing; 18 – cooling jacket (adapted from [12, 14])

The oil supply valves to the working cylinder are installed on the sleeve and serve to dose the oil to the surface of the cylinder mirror. In the system under consideration, each valve is equipped with

For preliminary review

a spring pressure accumulator (Fig. 6.12 *f*), the purpose and operating principle of which will be discussed below.

In addition to systemswith mechanical drive of pumping elements, manufacturers of marine engines and components for them have developed pump modules with hydraulic drive and electronic flow control. Such systemsare best suited for integration into electronic engine control systemsand at the same time suitable for use in traditional mechanically controlled engines. The general arrangement of the elements of such a system developed by SKF Lubrication SystemsGermany GmbH and designated CLU4-C is shown in Fig. 6.13. The main element of such a system is the hydraulic lubricating oil supply module, one of the options is shown in Fig. 6.14. The goal of developing CLU 4 was to bring oil consumption into line with the main engine operating modes and operating conditions. The main factors that are taken into account during system operation include: engine speed, load, technical condition, quality and composition of fuel and lubricating oil.

As can be seen in Fig. 6.13, the main elements of the CLU 4 system are;

- hydraulic oil supply modules with electronic control, each of which serves a separate engine cylinder and is designed to connect from 3 to 12 oil valves;
- oil valves installed on the cylinder liner;
- a pump unit supplying oil to the system;
- control units, both individual modules and the entire system as a whole;
- a reservoir filter with a scale for measuring the total oil consumption.

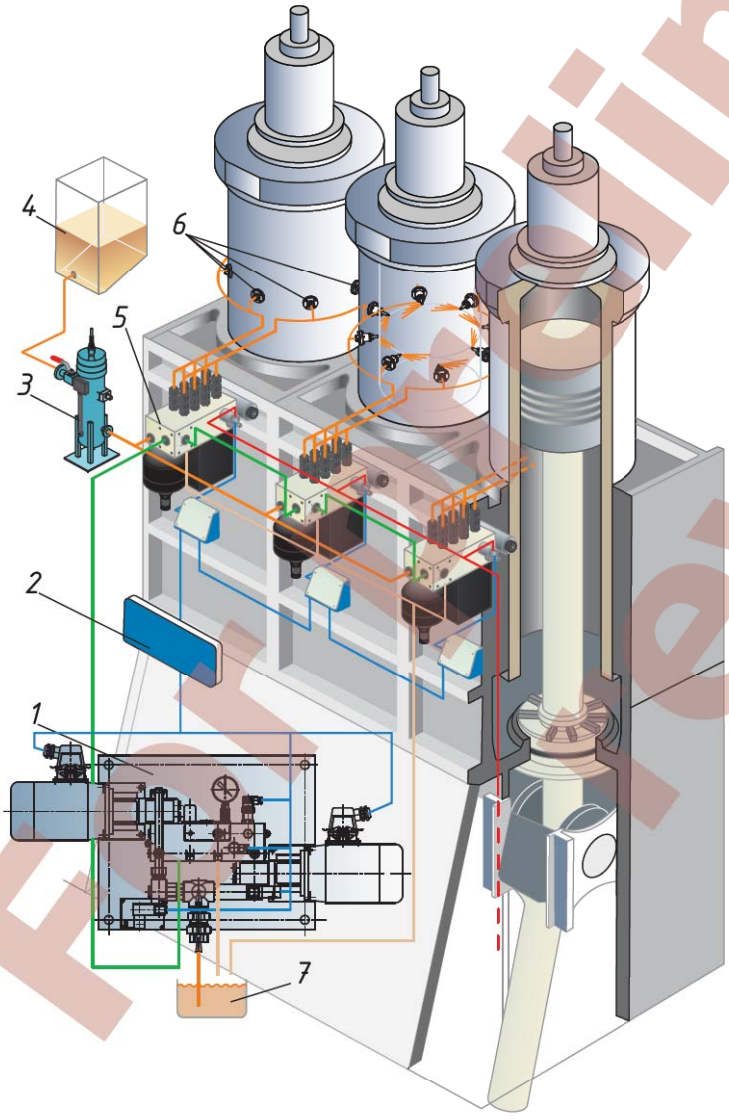


Figure 6.13 – Main elements of the working cylinder lubrication system: 1 – control oil supply module; 2 – electronic control unit; 3 – cylinder oil filter; 4 – cylinder oil supply tank; 5 – hydraulic cylinder oil supply unit; 6 – nozzles for supplying oil to the working cylinder; 7 – tank for hydraulic oil (adapted from [15])

Figure 6.14 – General view of the hydraulic cylinder oil supply unit (adapted from [16])

Similar systemshave been developed by other marine engine manufacturers. For example, the working cylinder lubrication system developed by MAN for mechanically controlled engines, called Alpha lubricators.

Preliminary review

All of these lubrication systemsserve to supply a strictly defined volume of oil into the cylinder through several injectors, not at every engine revolution, as was implemented in mechanical engine-driven systems, but once every few revolutions. In this case, both the oil supply frequency and its quantity are determined by the electronic control module and depend on the engine operating mode. Precise synchronization of the supply modules with the position of the piston in the working cylinder ensures that all cylinder oil is supplied directly to the piston ring pack there.

The general structure and operating principle of the hydraulic supply module is shown in Fig. 6.15.

The pump unit supplies lubricating oil to the hydraulic modules at a pressure of 4.5 MPa. The module consists of several oil supply cylinders, made in a common housing as axial holes. Each sleeve houses a separate plunger driven by a common drive collar, made integral collar the hydraulic drive piston. When ready to supply oil, the hydraulic piston of the drive and all plungers are in the extreme right position (Fig. 6.15 *a*). In this case, through a radial groove on the block housing, all cyl-inders communicate with the internal cavity of the module, filled with cylinder oil, which enters the cavity of each of them. When an electrical signal is received, the electromagnetic control valve opens the oil supply to the hydraulic cylinder of the drive. Under the influence of an oil pressure of 4.5 MPa, the hydraulic piston moves the drive collar along with the plungers, which block the filling groove, and the remaining oil is forced into the pressure lines connected to the fittings installed on the hydraulic module cover. The maximum stroke of the drive piston, and, consequently, the maximum supply of each plunger is adjusted using a thrust screw installed along the axis of the drive cylinder. Thus, the drive side uses an accumulator oil supply system from a pressure line, and the injection side is a high-pressure direct oil displacement system. The return of the plungers and hydraulic piston to their original state is carried out under the action of a spring, after the electrical signal from the control valve is removed and the spool pair connects the cavity of the hydraulic cylinder with the drain line.

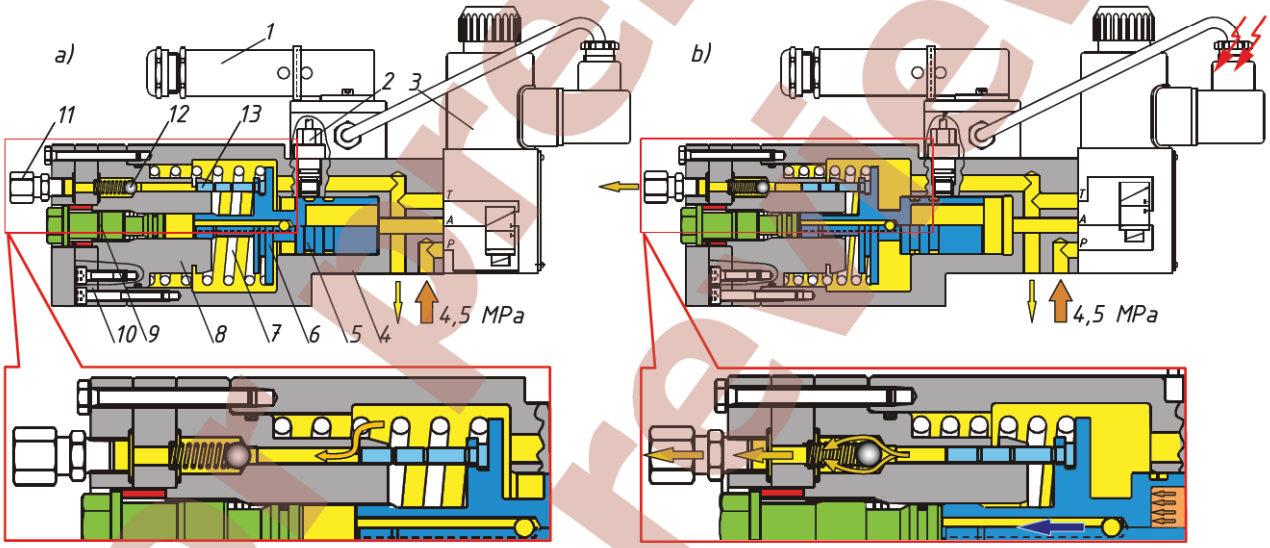


Figure 6.15 – General structure of the cylinder lubricator of low-speed engines of the MC-C series from MAN: *a* – filling the discharge cylinders; *b* – oil injection; 1 – electrical connector; 2 – displacement sensor; 3 – electrohydraulic valve; 4 – housing; 5 – hydraulic piston; 6 – plunger drive collar; 7 – return spring; 8 – cylinder block; 9 – maximum supply adjustment screw; 10 – cylinder block cover; 11 – connecting fitting; 12 – ball valve; 13 – plunger (adapted from [17])

To stabilize the operating pressure in the drive system, the pressure line of each module is equipped with a diaphragm-type gas pressure accumulator. This design ensures the same amount of lubricating oil is supplied to each oil supply valve and provides maximum resistance to clogging of individual lubricating oil nozzles.

On larger bore engines, each cylinder has two hydraulic modules (each serving half the lube oil

For preliminary review

injectors), while small bore engines (with fewer lube oil injectors per cylinder) are served by one module per cylinder.

On engines with dual-row oil supply valves, if one hydraulic module fails, the oil supply from the other module is automatically doubled and an alarm is generated. For engines with single-stage valves, an alarm is generated and the engine is stopped.

An inductive sensor installed in each module monitors the movement of the hydraulic piston. The signal is transmitted to the control unit. This ensures feedback between the actuator and the control system.

The pumping station includes two pumps (one operating, the other standby with automatic start), a filter and coolers.

When such cylinder lubrication systemsare used to complete mechanically controlled engines, a separate electronic system is installed on them, consisting of a main computer that controls normal operation, a switching unit and a backup unit. To synchronize the oil supply, crankshaft position sensors are additionally installed on the engines.

In electronically controlled engines, these functions are integrated into the engine management system.

An example of such a module integrated into the general engine control system of the ME series from MAN is shown in Fig. 6.16.

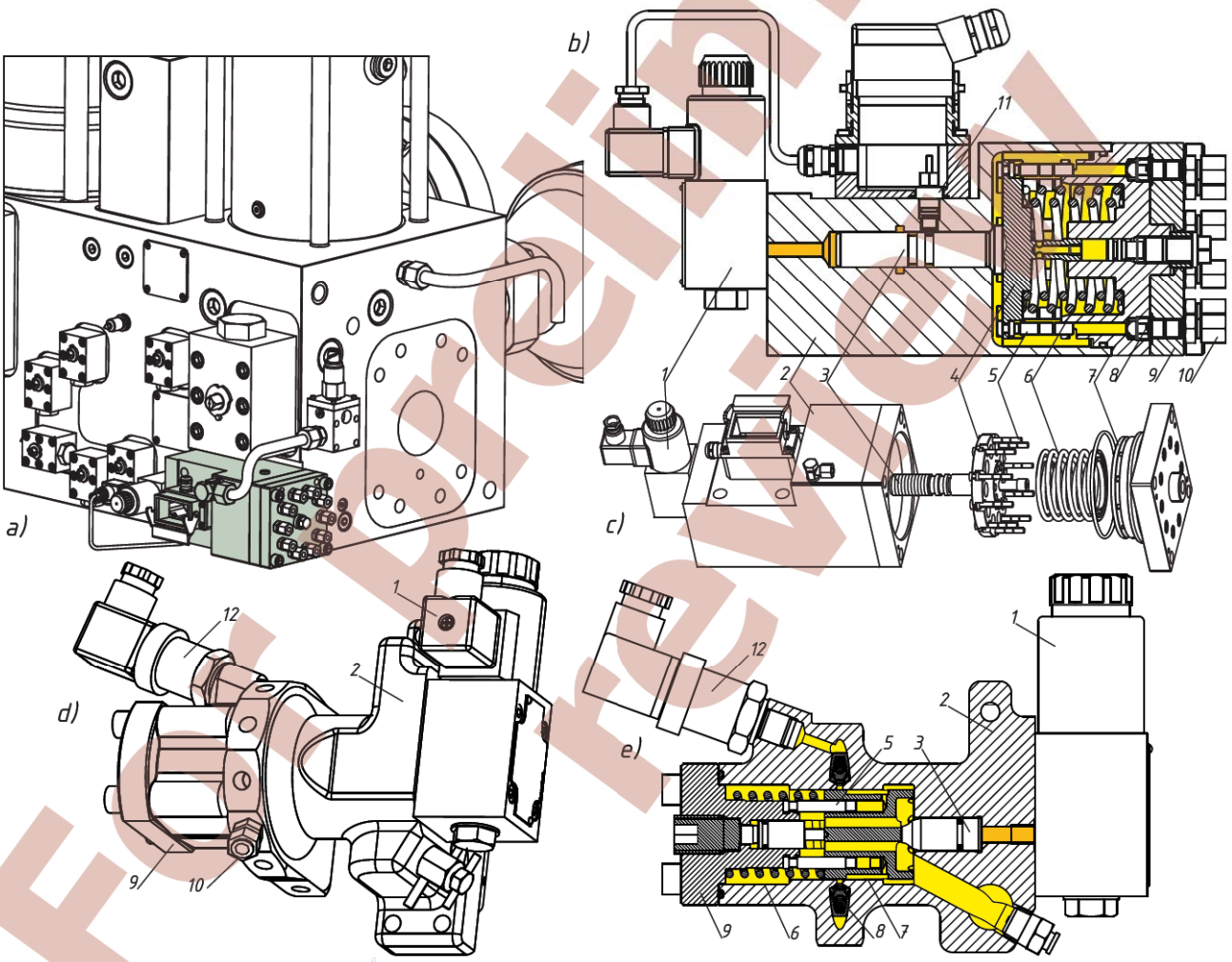


Figure 6.16 – Location of the Cylinder Lubricator on the hydraulic block of MAN ME series engines (*a*), its structure (*b*) and main elements (*c*). General view (*d*) and structure (*e*) of the hydraulic module for supplying cylinder oil to X series engines, WinGD: 1 – solenoid valve; 2 – housing; 3 – hydraulic piston of the drive; 4 – plunger drive collar; 5 – plunger; 6 – return springs; 7 – cylinder block; 8 – check valves; 9 – cover; 10 – connecting fittings; 11 – drive displacement sensor; 12 – pressure sensor (adapted from [11, 18])

For preliminary review

Structurally, the oil supply module does not differ from the devices we previously reviewed. A distinctive feature is that the hydraulic drive uses control oil from the common hydraulic engine control system. Let us remember that the pressure in this system is about 20 MPa. In this regard, the plunger drive piston has a significantly smaller diameter.

Direct installation of the module on the hydraulic control unit for fuel injection and gas distribution (Fig. 6.16 *a*) makes it possible to avoid installing additional pressure accumulators, since these devices are available on each unit, due to which the pressure in the system is maintained at a fairly stable level.

The cylinder oil supply modules used on X-series engines from WinGD have a similar device (Fig. 6.16 *d*). The only fundamental feature is that in this design the moving element is a cylinder block driven by a hydraulic piston. In addition to the discharge function, the cylinder block also performs the function of a spool. At the initial stage of its movement, it cuts off the filling cavity from the feed cylinders and connects them to the pressure lines.

As noted, the physical and chemical processes in the working cylinder differ significantly from those that operate in the crankcase space, therefore cylinder oils perform a number of additional functions, which include:

- lubrication of friction surfaces and ensuring a stable hydrodynamic oil film between the liner and piston rings;
- sealing the gas joint between the set of piston rings and the cylinder liner;
- removal of combustion products, pyrolysis of lubricating oil and fuel, wear products from friction surfaces (detergent properties of oil);
- neutralization of acids formed during combustion to reduce the impact of cold corrosion on the surface of the liner.

Depending on which factor is predominant under a given operating condition, the required oil flow rate may vary significantly. So, for example, when running in a cylinder after replacing any elements of the cylinder-piston group, the supply should be increased for more effective lubrication of friction pairs and removal of wear products. When operating on fuels with a high sulfur content (if, for example, the ship is equipped with an exhaust gas purification system), the supply of cylinder oil must be sufficient to neutralize the resulting acids due to the alkaline components of the oils. In general, for modern low-speed marine engines, the specific consumption of cylinder oil lies in the range of 0.6...1.35 g/(kWh) and can vary significantly depending on the load, the type of fuel and oil used.

The consumption of cylinder oil is significantly affected by the method of its supply to the surface of the working cylinder, which, in turn, depends on the type of devices used for introducing oil into the working space of the engine, and the method of its distribution over the friction surfaces.

The most common method of oil distribution is shown in Fig. 6.17 *a*, *b*.

Oil is supplied by special valves, the tips of which are installed in the upper part of the cylinder liner and go into special oil distribution grooves cut on the working surface of the liner. Along these grooves, the oil spreads around the perimeter of the bushing and, being picked up by a set of piston rings, is distributed over its entire surface. Oil supply valves are installed in radial holes in the bushing and connected to the oil supply module (or progressive distributor) by tubes equipped with appropriate fittings (Fig. 6.17 *c*).

The simplest oil supply valve consists of a housing, a tip and a check ball valve installed between them (Fig. 6.17 *d*). The valve protects the oil line from gases from the working cylinder when the pressure in it is higher than the pressure in the oil line.

When using the simplest valves as part of mechanically driven oil systems, they have a significant drawback. As the engine rotation speed increases, the speed movement of the plunger increases, as a result of which part of the supplied oil is torn off the surface of the tip and injected into the working cylinder without getting onto the walls of the sleeve. Most of this oil settles on the piston surface and burns, resulting in the need to increase the oil flow rate to compensate for losses.

The use of the simplest valves as part of systemswith electronic synchronization makes it possible to eliminate the described disadvantage, since the oil supply occurs when the piston rings pass

For preliminary review

along the valve tip and all the oil enters the gaps between the rings and the piston.

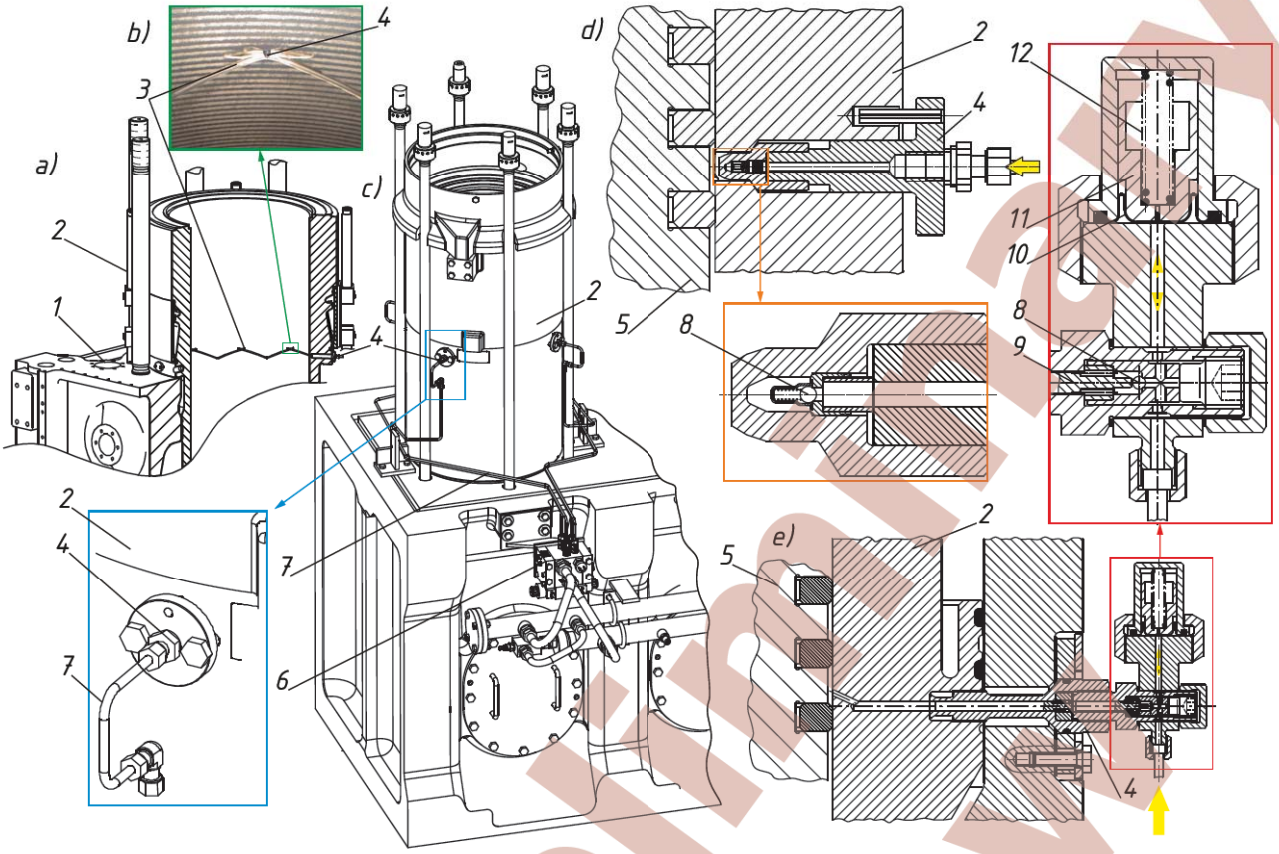


Figure 6.17 – Location of the main elements of the lubrication system for working cylinders and the main types of valves for supplying oil to the working surface of the cylinder liner: *a* – location of the oil distribution grooves on the working surface of the cylinder liner; *b* – general view of the oil distribution grooves; *c* – location of the hydraulic oil supply module, oil lines and oil supply valves on the X-40 engine from WinGD; *d* – oil supply valve for the X-40 engine from WinGD; *e* – oil supply valve with spring accumulator for Sulzer RTA series engines; 1 – cylinder block; 2 – cylinder liner; 3 – oil distribution grooves; 4 – oil supply valve; 5 – piston; 6 – hydraulic module for supplying cylinder oil; 7 – oil lines; 8 – ball valve; 9 – rod; 10 – separating diaphragm; 11 – accumulator piston; 12 – accumulator spring (adapted from [11, 19])

To prevent pulsed oil supply, some manufacturers equip each oil supply valve with a piston pressure accumulator. The design of such a valve is shown in Fig. 6.17 *f*. The oil supplied by the pump section enters the valve cavity, from where it enters the gap between the internal channel and the rod placed in it through the ball valve. Moving along the gap, the flow speed slows down, and oil enters the cylinder at a speed insufficient to separate it from the nozzle hole. At moments when the oil injection rate exceeds the capacity of the oil outlet channel for cylinder lubrication, excess oil enters the spring accumulator, overcoming the spring force, lifting the piston and deforming the separating diaphragm. This way the accumulator is charged. After the oil supply from the discharge element stops, the accumulator gradually displaces the accumulated oil through the nozzle hole onto the surface of the cylinder liner. Naturally, the feeding process is only possible when the oil pressure in the valve cavity is higher than the pressure in the working cylinder and the rod does not hold the ball valve closed. Thus, oil supply begins automatically as soon as the pressure at the nozzle exit is lower than the lubricating oil pressure, for example, when passing a set of piston rings.

A fundamentally different approach in the field of lubrication of working cylinders is to supply oil to the working surface of the bushing under high pressure in the form of a finely sprayed oil aerosol (Fig. 6.18 *a*).

This technology was developed by Hans Jensen Lubricators and was called SIP, which is an ab-

For preliminary review

breviation for Swirl Injection Principle.

To implement pressure lubrication, supply valves are used, which functionally have much in common with fuel atomization nozzles. The general view of such a valve is shown in Fig. 6.18 *b*, and its general structure is shown in Fig. 6.18 *c*.

Oil is supplied to the valve at a pressure of 3.8…4.2 MPa. In this case, plunger units (Fig. 6.18 *d*) that have proven themselves in operation or specially designed multi-piston pumps with a hydraulic drive are used as supply pumps.

The operation of the pumping units is synchronized with the operation of the engine, so that oil is injected into the cylinder at the beginning of the compression stroke, when the pressure in it is significantly less than the pressure of the supplied oil.

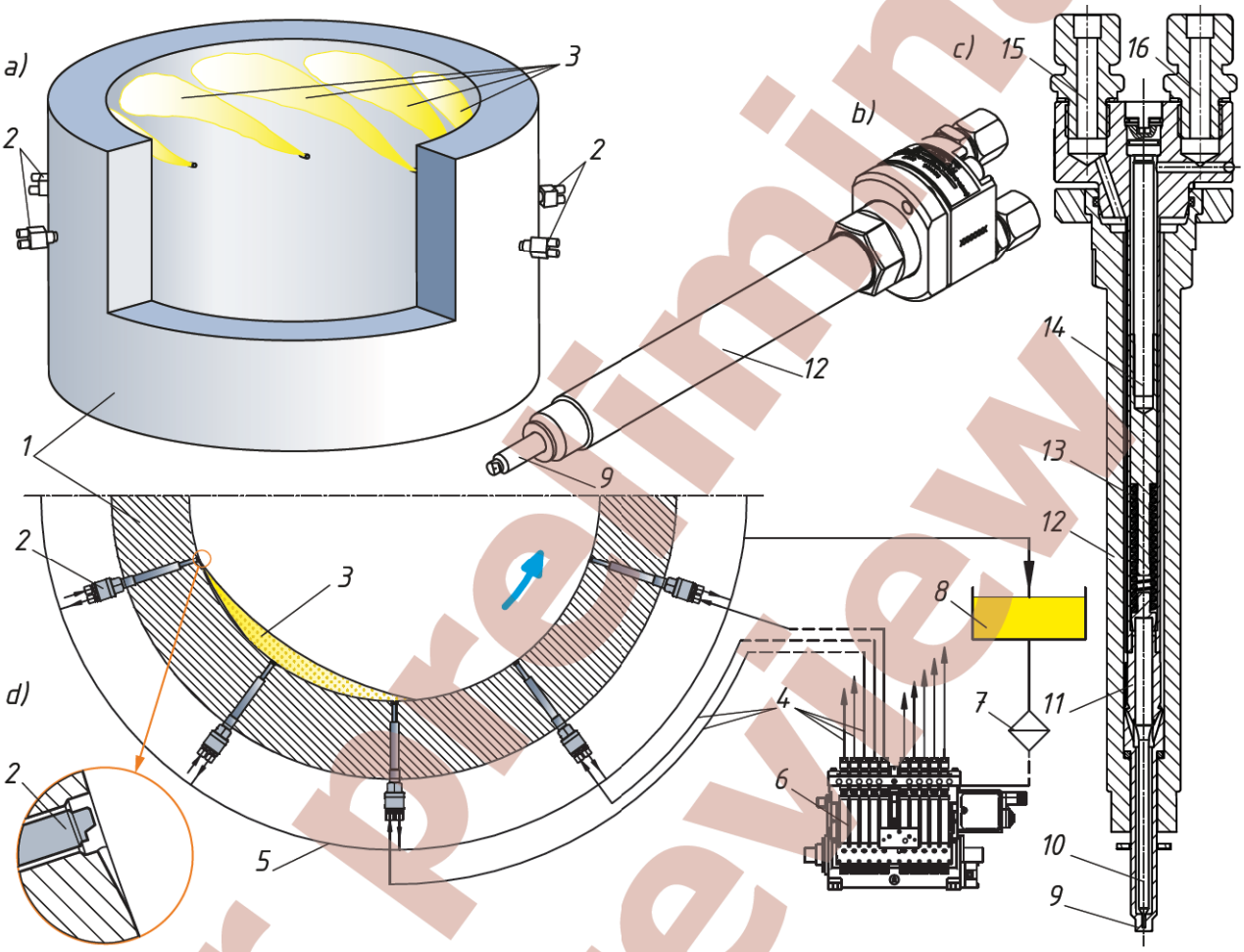


Figure 6.18 – High pressure lubrication system for working cylinders, type HJ SIP. *a* – distribution of oil aerosol along the wall of the working cylinder; *b* – general view of the oil supply valve; *c* – oil supply valve device; *d* – general system diagram; 1 – cylinder liner; 2 – oil supply valve; 3 – aerosol cloud; 4 – oil supply lines; 5 – leakage drainage line; 6 – oil pump; 7 – cylinder oil filter; 8 – cylinder oil supply tank; 9 – spray tip; 10 – needle valve; 11 – needle valve guide; 12 – valve housing; 13 – spring; 14 – adjusting screw; 15 – pressure line connection fitting; 16 – drain line connection fitting (adapted from [13])

Through tangentially directed holes in the tip of the nozzle, a stream of oil is injected along the cylinder wall and atomized into a fine aerosol, which settles evenly on the surface of the upper part of the bushing. The uniformity of distribution is also enhanced by the tangential air flow passing through the cylinder. The air picks up oil droplets and carries them along the wall, and the resulting centrifugal force presses them to the surface of the bushing, causing them to settle in the form of a thin, uniform film.

As a result, in the upper part of the bushing, where oil is most needed to reduce friction, neutral-

For preliminary review

ize acids and remove combustion products, a uniformly lubricated surface is formed, prepared for the passage of piston rings. With the downward movement of the piston and piston rings, oil is distributed over the remaining working surface of the cylinder liner.

This technology is used both in the construction of new engines and for the modernization of those already in operation. Its use can significantly reduce cylinder oil consumption without compromising engine reliability, improve its environmental performance and increase the time between cleanings and repairs.

Systems similar to the method of applying an oil film to the surface of the cylinder liner have been developed by other manufacturers of marine engines. Thus, the WinGD company developed and implemented a system called Pulse Jet, in which the cylinders are lubricated by an electronically controlled accumulator system. This technology is combined with the use of traditional oil distribution grooves.

6.4 Turbocharger lubrication systems

Turbochargers are usually lubricated with oil from the engine's circulating lubrication system. However, some types of turbocharger units have their own lubrication system.

An example of such turbochargers are units of the VTR series from ABB (Fig. 5.59). The rotor bearing units of these units are placed in separate crankcases, which are used as reservoirs for storing oil reserves. In small turbochargers of this series, oil pressure discs are used to supply oil to the bearing assembly (Fig. 6.19 *a*), and in large ones, gear pumps mounted on the rotor shaft are used (Fig. 6.19 *b*).

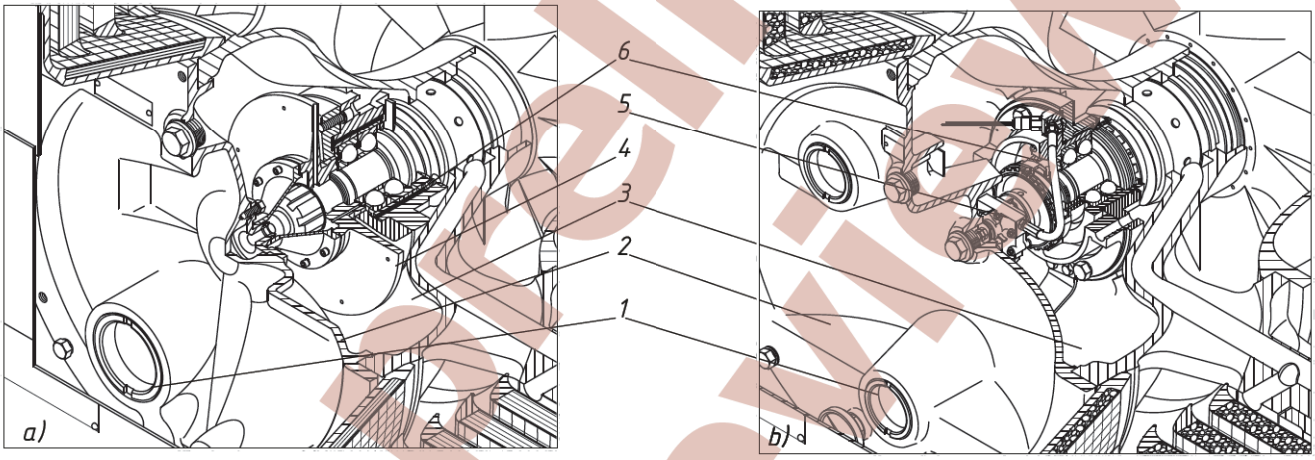


Figure 6.19 – Crankcase bearing unit for VTR series turbochargers from ABB with a pressure disc (*a*) and a mounted oil pump (*b*) on the compressor side: 1 – port for monitoring the oil level in the crankcase; 2 – crankcase cover; 3 – crankcase; 4 – pressure disc; 5 – mounted oil pump; 6 – bearing unit (adapted from [20])

The pressure disc operates as a centrifugal supercharger, which draws oil from the lower part of the crankcase, which acts as a reservoir, and distributes it along a special supply channel through radial drillings along the inner surface of the crankcase. As it drains, some of the oil enters the bearing assembly, lubricating it. For more loaded turbochargers, a mounted gear-type oil pump is installed on the rotor axis; it sucks oil from the crankcase and delivers it through a special tube to the bearing unit, ensuring its effective lubrication and cooling. In both cases, the crankcase covers are equipped with ports for monitoring the oil level and plugs for filling and draining it.

The main advantage of individual lubrication of turbochargers is its autonomy. In the event of an engine shutdown, especially an emergency one, the turbocharger will continue to rotate by inertia for quite a long time, during which lubricant will continue to be supplied to the bearings.

For turbochargers with bearings located in the middle housing, lubricant is supplied, as a rule, from the engine circulation system through a pipeline on which a pressure reducing valve is in-

For preliminary review

stalled that maintains the required lubricating oil pressure. If necessary, the supplied oil is additionally filtered. To prevent oil from draining back into the system when the engine is not running, a check valve is installed. Lube oil pressure is monitored behind the check valve using a pressure switch and pressure gauge

The prepared and cleaned oil enters the bearing housing, filling the radial and axial clearances and holes in the bearings through the annular channels. Passing through the gaps in the bearings, the oil lubricates and cools friction pairs and removes wear products. Flowing out of the gaps, oil splashes along the walls of the bearing assembly, flows to its lowest point, from where it returns through the drain pipeline to the crankcase or service tank of the engine. The diameter and installation angle of the drain pipe must ensure direct drainage of all oil at the maximum angle of inclination for which the installation is designed. To avoid excess pressures that could disrupt the operation of the emergency lubrication system, a ventilation system is provided in the bearing unit housing.

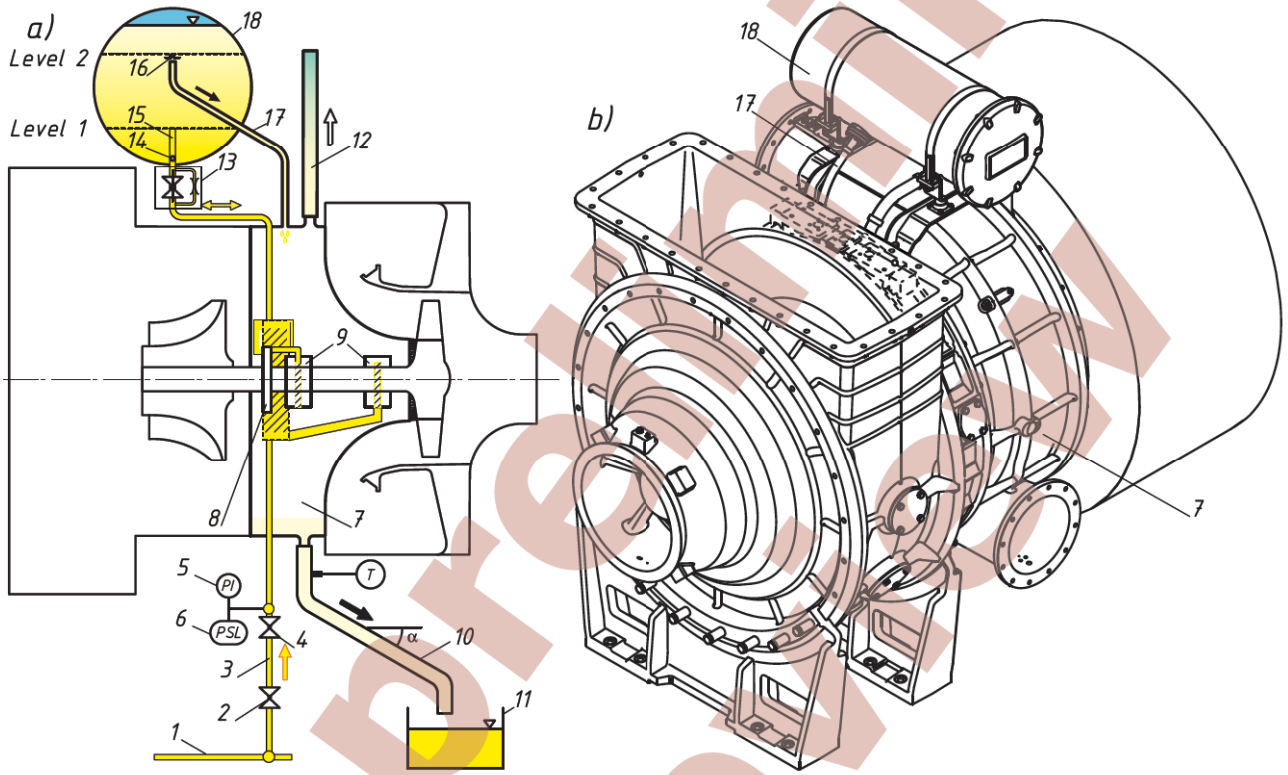


Figure 6.20 – Diagram of the lubrication system for MAN TCA series turbochargers (*a*) and the location of the emergency lubrication system oil tank on the turbocharger housing (*b*): 1 – Oil supply from the engine lubrication system; 2 – pressure reducing valve; 3 – oil supply pipe to the turbocharger; 4 – check valve; 5 – pressure switch; 6 – pressure gauge; 7 – bearing housing; 8 – thrust bearing; 9 – radial bearing; 10 – drainage pipe; 11 – service tank or crankcase; 12 – ventilation pipe; 13 – check valve with a calibrated bypass channel; 14 – calibrated drain hole; 15 – supply (drain) pipe; 16 – throttle insert; 17 – overflow pipe; 18 – reserve tank (adapted from [21])

An emergency lubrication system is installed on the turbocharger and ensures lubrication of the bearings after the engine is stopped. The oil reserve in the system reservoir should ensure its supply at least until the rotor frequency decreases to 20% of the nominal one.

The system is filled through a calibrated bypass channel in the check valve housing. The valve itself remains closed while the engine is running.

As long as the lubricating oil level does not reach the end of the overflow tube, there is no pressure in the emergency lubrication system.

A throttle insert is installed in the overflow tube channel, which is adjusted so that more oil flows into the storage tank than flows out of it. Therefore, after the oil reaches the cut-off level of

For preliminary review

the overflow tube, the tank continues to fill. The air in its upper part begins to compress, the pressure in the tank increases, and the drainage speed increases. This continues until a balance is established between the inflow and outflow of oil from the reserve tank. In this case, a certain excess pressure is established in the system, which is constantly maintained during engine operation.

When the engine stops, the pressure in the lube oil system decreases. In this case, the check valve at the inlet to the engine oil system closes, and the check valve on the reserve tank opens. The oil, displaced by compressed air, begins to flow under pressure from the emergency lubrication system back into the turbocharger to the bearing bushings. When the lubricating oil level drops below the overflow tube (level 2), the excess pressure in the system ceases, but oil continues to flow to the turbocharger bearings by gravity at a reduced flow rate. As soon as the height of the supply (drain) pipe is reached (level 1), the remaining volume in the tank flows out through a small calibrated hole in its lower part. This ensures three-stage lubrication of the turbocharger bearings during the run-down of its rotor. The most intense pressure is at the moment the engine stops, gravitational pressure is of medium intensity when the speed has dropped significantly and low intensity when the rotor has practically stopped.

6.5 Friction and lubrication modes

In friction pairs between individual engine structural elements, various friction and lubrication modes can occur depending on the speed and method of relative movement of the rubbing surfaces, the loads acting on them, the amount and method of oil supply. There are three main modes of friction (lubrication):

- hydrodynamic;
- contact-hydrodynamic;
- borderline.

Hydrodynamic or liquid mode is a lubrication method in which the thickness of the oil layer is sufficient to prevent direct contact of micro-irregularities of the rubbing surfaces. If the surfaces are separated by a relatively thick layer of oil (more than 0.1 μm), then the properties of the oil in the film and in the volume are the same.

In the absence of rotation, the shaft, under the influence of loads and its own weight, occupies an eccentric position, and on both sides of it a wedge-shaped gap is formed with a maximum gap in the upper part Δ (Fig. 6.21 *a*).

When the shaft rotates, the layer connected to its surface by sorption forces carries along subsequent layers and forces them into the narrowed part of the wedge-shaped gap. The mutual connection between the film adsorbed on the surface of the shaft and the oil in the gap is due to viscous forces (Fig. 6.21 *b*). Thus, the oil is involved in a relative movement in the direction of rotation and in the longitudinal direction towards the ends of the bearing. The limited clearances prevent the free flow of oil, which is practically incompressible under these conditions. Therefore, hydrodynamic pressure is created in the gap, acting on the shaft, which increases in the direction of decreasing the size of the slit along the circumference. In the longitudinal direction, the pressure diagram has the form of a hyperbola (Fig. 6.21 *c*). As a result, the shaft is raised and shifted in the direction of rotation (Fig. 6.21 *b*) by an amount *e*. Some of the oil flows out through the ends of the bearing, and the rest is pumped through the narrow part of the gap. In this case, the shaft is raised by an amount h_{min} until an equilibrium state occurs, at which the flow area will be sufficient to pass the remaining part of the oil.

The hydrodynamic pressure developed in the lubricant layer ensures the separation of the shaft and bearing surfaces, and the friction between the surfaces is replaced by internal friction between the oil layers.

As the rotation speed increases, the hydrodynamic pressure and the thickness of the oil film increase, the shaft floats up, and the friction force quickly decreases. With a further increase in rotation speed, the bearing capacity of the oil layer increases, but at the same time the forces of liquid friction in the layer increase.

For preliminary review

Oil viscosity significantly affects the formation of a hydrodynamic film, but as it increases, the energy costs for friction work also increase and heat generation in the bearings increases.

The coefficient of friction under hydrodynamic lubrication conditions is extremely low, and the absence of direct contact between the rubbing surfaces minimizes their wear. In practice, under engine operating conditions, frequent violations of the purely hydrodynamic lubrication regime are inevitable. They can occur when starting the engine, sudden load fluctuations, or solid particles entering the gap whose size exceeds the thickness of the hydrodynamic film. The cause of wear during hydrodynamic lubrication is also electrostatic discharges that occur during friction between surfaces.

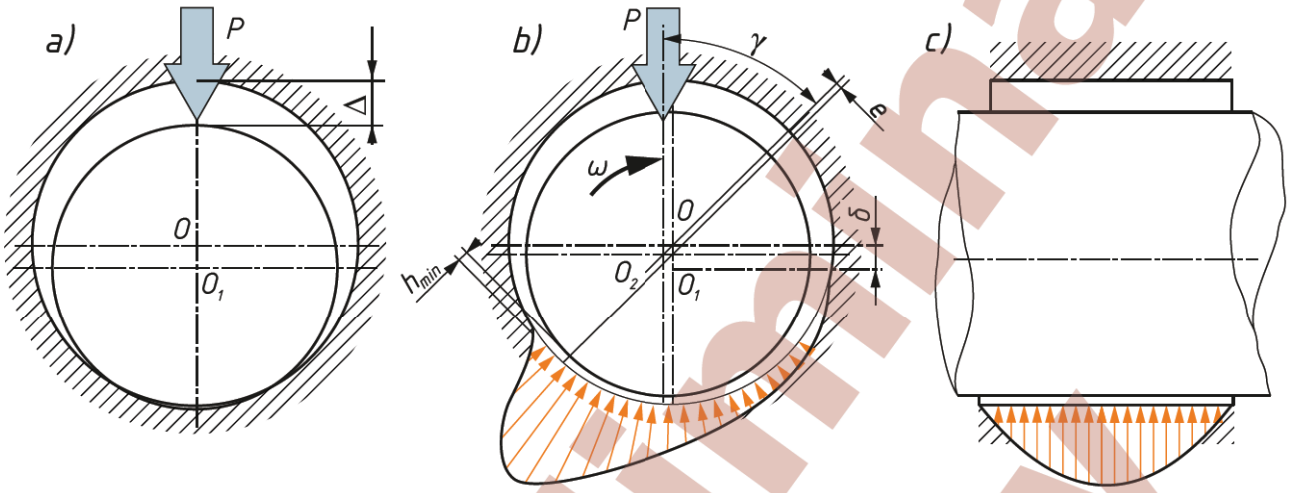


Figure 6.21 – Schemes formation of a hydrodynamic layer of oil (oil wedge) separating the friction surfaces (*a*, *b*) and pressure distribution diagrams in the bearing (*b*, *c*) (adapted from [22])

Contact-hydrodynamic (elastohydrodynamic) and hydrostatic lubrication are typical for friction pairs that, in the presence of oil, contact at a point or along a line. Rolling bearings, heavily loaded gear teeth, and cam mechanisms operate under such conditions. These elements are characterized by high contact loads under deformed film conditions. At the same time, the molecular structure of the oil undergoes changes, the viscosity of the oil increases many times, and it loses the properties of a Newtonian fluid. At the same time, plastic deformation of the metal occurs under the layer of deformed oil.

A similar lubrication regime is typical for crosshead bearings of low-speed engines. The mutual movement of friction pairs in these bearings occurs under conditions of heavy loads in a small arc and is reciprocating in nature, at relatively low speeds, insufficient for the occurrence of hydrodynamic friction. Therefore, to maintain the oil layer in crosshead bearings, increased supply pressure of lubricating oil is used, which is supplied to the friction zone through a developed system of oil distribution grooves.

Boundary lubrication regime. Deterioration of the conditions for retaining oil in the friction zone, for example, with increasing gaps, as well as increased pressure and temperature, contribute to a decrease in the thickness of the separating oil film. When the layer reaches a thickness of 0.02...0.1 μm, the liquid lubrication regimes are disrupted. Under such conditions, the friction coefficient does not depend on the bulk viscosity of the oil, but on the presence of polar-active components in it, forming a thin separating (boundary) film on the surfaces.

Films that form on surfaces can be of chemical (chemisorption) and physical (adsorption) origin. Oxide films that are formed as a result of the interaction of the friction surface material with oxygen or additives contained in the oil are films of chemical origin. They firmly adhere to surfaces and are destroyed only when the melting temperatures of the oxides are reached. Films of chemical origin also include various soaps formed by the interaction of higher organic acids in oil with metal. The presence of oxide films helps prevent seizures that can occur during direct contact of metal surfaces.

Adsorption layers are formed on friction surfaces by polar-active hydrocarbons, lining up per-

For preliminary review

pendicular to them in the form of chains (Fig. 6.22). The surface layers are several molecules thick, with each molecule attached at its polar end to the non-polar end of the previous molecule. The further from the surface, the weaker the connection between the molecules, so after several layers their arrangement is chaotic. The junctions between layers of oriented and randomly located molecules are planes of facilitated sliding.

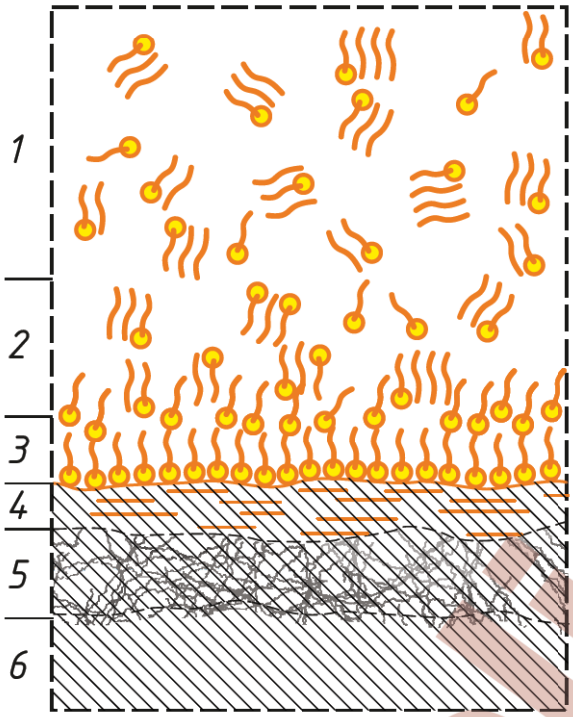


Figure 6.22 – Scheme of the structure of boundary layers: 1 – liquid; 2 – boundary phase; 3 – adsorbed layer; 4 – chemical compounds (chemisorbed boundary film); 5 – zone of deformed metal; 6 – metal

Boundary films can withstand pressures of thousands of megapascals, but they are easily destroyed when the temperature rises, since this increases the thermal movement of molecules, which destroys polar bonds. The orientation of the molecules is disrupted, and the boundary film loses adhesion to the metal surface. The temperature at which this occurs is called the desorption temperature. For mineral oils it is 90...100°C.

Polar molecules contain not only fatty acids, but also oil oxidation products, organic acids formed during the operation of oil in the engine. This explains, to a certain extent, the improvement in the lubricating properties of oils during the aging process.

To enhance the polarity of molecules, 0.5...2% of the most stable fatty acids, castor oil, amines and esters, etc. are specially introduced into base oils.

Special extreme pressure additives are added to oils intended for severe operating conditions – compounds of sulfur, chlorine, and, less commonly, bromine, iodine and phosphorus. When they react with metal, thin films of iron sulfide, chloride or phosphide are formed on its surface, separating the rubbing surfaces and significantly reducing wear. The most effective extreme pressure agents are additives based on sulfur compounds; they are effective in conditions of boundary friction at temperatures up to 650°C.

Literature used when writing section 6:

1. Operating Instructions TBD 645. 0299 6391 en, Motoren-Werke Mannheim AG, Mannheim. Germany. – 244 p.

2. General Engine operating instructions Typ M20. Caterpillar Motoren GmbH & Co. KG. 2005. – 2150 p.

3. Technical Documentation Operation, Maintenance, Mounting and Disassembly Instructions for the Series L3NG Screw Pump. Leistritz Pumpen GmbH – Technische Dokumentation 200 KB 17858 00100. 2009. – 38 p.

4. Engine type W46B Instruction Manual. Wartsila Finland Oy Vaasa Factory. Finland. 2011. –

For preliminary review

458 p.
5. Instruction manual oil heater (U–Tube type). DongHwa Entec., 1575–6, Song Jeong Dong, Gangseo-GU, Busan, Korea. – 14 p.
6. Instruction Manual. Plate Heat Exchangers: M3 M6 M10 TS6 T2 T5 TL3 TL6. Alfa Laval. Part number 1644725-01., 2009. – 47 p.
7. Wärtsilä 18V34SG. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 416 p.
8. BOLL Automatic Filter Selfclean Type 6.46. Efficient, economical. BOLL & KIRCH Filterbau GmbH. Kerpen. – 7 p.
9. Spare parts catalogue. Engine type 18V32LN. Wärtsilä Finland Oy. Vaasa Factory. Finland. – 452 p.
10. Wärtsilä 32 Technology review. W-P0304. Wärtsilä Finland Oy. Vaasa Factory. Finland. – 16 p.
11. Engine Type S50ME-C8. Volume III Components Description. DS 42014238. STX Dalian Engine Co.,Ltd. – 202 p.
12. RTA72U-B. Operating Manual «Marine». Wärtsilä Switzerland Ltd, Switzerland. 2000. – 277 p.
13. HJ Sip Lubrication System (Patented). Hans Jensen Lubricators A/S. Hadsund. 2007. – 169 p.
14. Cylinder Lubrication System CLU 3 for 2-Stroke Crosshead Large Diesel Engines. Willy Vogel AG. Berlin, Germany. 2004 – 5 p.
15. Pulse Lubrication System for cylinder lubrication on large 2-stroke crosshead engines CLU4-C. Assembly Instructions according to EG RL 2006/42/EG. SKF Lubrication Systems Germany GmbH. Berlin, Germany. 2006 – 56 p.
16. Wärtsilä Pulse Lubricating System. Wärtsilä Corporation. 2007 – 4 p.
17. Lubricator System for Cutting the Cylinder Oil Bill and Reducing Emissions on MAN B&W Two-Stroke Engines. MAN B&W Diesel A/S. Copenhagen. 2001. – 4 p.
18. WinGD W-X82. Operation Manual «Marine». 2017 Winterthur Gas & Diesel Ltd. Winterthur, Switzerland. 2017. – 500 p.
19. Wärtsilä W-X40 Maintenance Manual «Marine». Document ID: DBAC412248. Wärtsilä Switzerland Ltd. Winterthur, Switzerland. 2013. – 438 p.
20. Extracts from ABB Turbocharging's Turbo Magazine 1990-2009 ABB Turbocharging Tips for the operator. ABB Turbo Systems Ltd. Baden, Switzerland. 2010. – 40 p.
21. Turbocharger Operating Instructions. Turbocharger type TCA66-40054. MAN Diesel. 2008 – 348 p.
22. Providing reliability of sliding bearings for gearwheels of high-loaded transport vehicles power transmissions during operation. / Savchuk V., Bulgakov N., Kuhtov V., Simahin A., Gritsuk I., Mateichyk V., Grascht R. // SAE Technical Paper 2018-01-0794, 2018, doi:10.4271/2018-01-0794.

For preliminary review

SECTION 7

Heat flows in marine engines. Heat removal systems from their structural elements

7.1 Thermal balance of marine engines

In internal combustion engines, only part of the heat in the process of converting thermal energy is converted into mechanical energy. For modern engines, this figure ranges from 38...55%; the rest of the heat is transferred to the environment in one way or another and represents irreversible heat loss. Part of the heat with exhaust gases is dissipated in the atmosphere, and part, by heat transfer through engine elements, is transferred to the cooling system and then to the sea water. A certain portion of the heat undergoes complex transformations, for example, part of the mechanical work is spent on overcoming frictional forces with subsequent conversion into thermal energy, which is also discharged into the cooling system. Some of the heat is returned to the cycle with air supplied from the turbocharger. The distribution of the heat released during fuel combustion in the engine Q_f into useful heat and various types of heat losses is called *external heat balance.* In Fig. 7.1 shows a diagram of the distribution of heat flows resulting from fuel combustion in a modern combined engine.

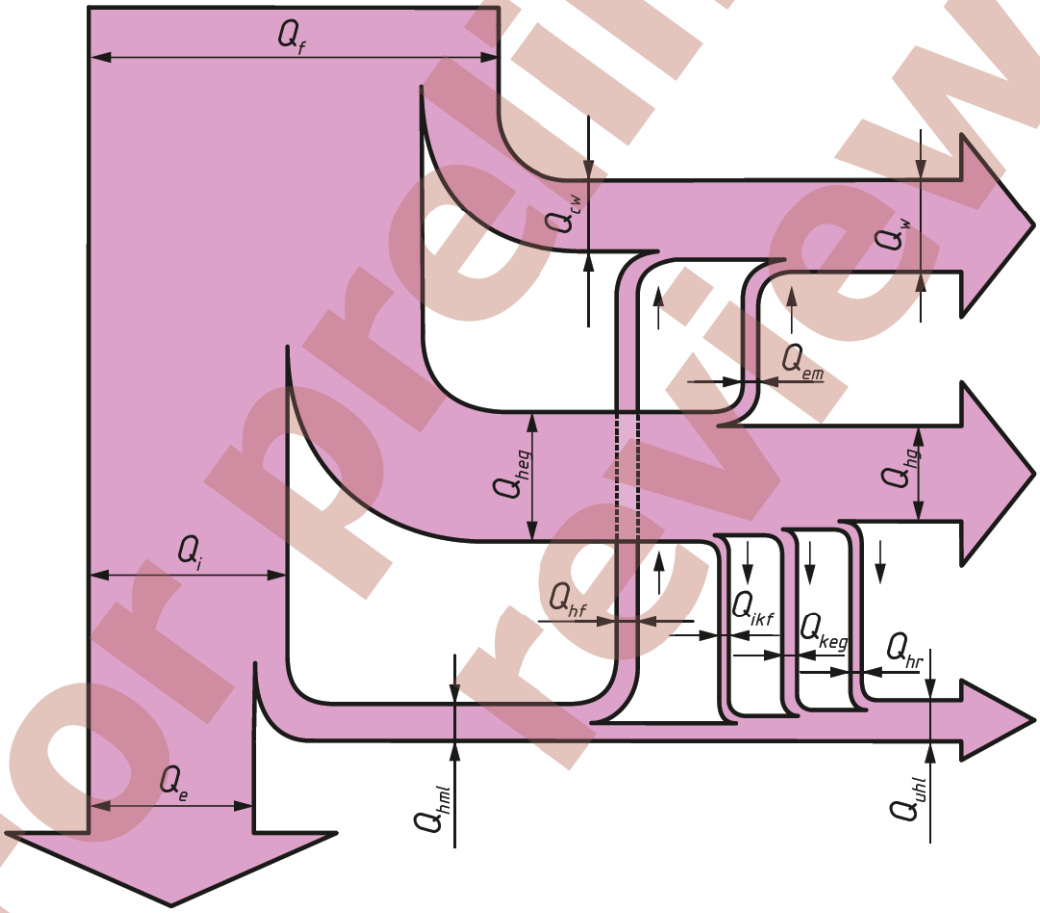


Figure 7.1 – Engine thermal balance diagram: Q_f – heat of combustion of fuel; Q_i – heat spent on indicator work; Q_e – heat spent on efficient work; Q_w – heat removal with coolant; Q_{cw} – heat loss due to heat exchange with the cylinder walls; Q_{hf} – heat loss from friction; Q_{em} – heat loss to cooling or the environment from the exhaust manifold; Q_{heg} – heat loss with gases after the cylinder; Q_{hg} – heat removed with exhaust gases; Q_{uhl} – total unaccounted heat losses; Q_{hml} – heat equivalent to mechanical losses; Q_{ikf} – heat loss from incomplete combustion of fuel; Q_{keg} – heat equivalent to the kinetic energy of gases; Q_{hr} – heat loss with radiation

For preliminary review

The external heat balance characterizes the degree of engine perfection and is the starting point for designing cooling systemsand determining rational ways to utilize heat losses. Due to the significant difficulties in calculating heat losses for individual components, the heat balance is determined experimentally at steady-state engine operating conditions.

In the most general form, the equations of the external heat balance of a diesel engine, estimating these losses in absolute units per 1 hour of operation (kJ/h), can be written as:

$$Q_f = Q_e + Q_w + Q_{hg} + Q_{uhl}. \quad (7.1)$$

where Q_f is the heat obtained as a result of fuel combustion in the engine cylinders; Q_e – heat equivalent to the effective operation of the engine; Q_w – heat removed from the engine to the cooling system; Q_{hg} – heat carried away with exhaust gases; Q_{uhl} – unaccounted heat losses.

The remaining losses, shown in the diagram in Fig. 7.1 can be introduced into these equations if a more detailed assessment is necessary.

The heat removed from the engine to the cooling system Q_w is often considered to be the sum of the heat removed from the circulating oil Q_{lo} and the heat removed directly into the cooling water Q_{cs} circulating in the engine cooling system.

For practical purposes, it is more convenient to present the heat balance in specific units kJ/(kWh) related to engine power:

$$q_f = q_e + q_w + q_{hg} + q_{uhl}.$$

Components of specific heat balance:

- heat supplied to the cylinder as a result of fuel combustion, mass m_f:

$$q_f = \frac{Q_e}{N_e} = \frac{m_f Q_l}{N_e} = g_e Q_l;$$

- heat converted into effective work on the diesel shaft:

$$q_e = \frac{Q_e}{N_e} = \frac{3600 N_e}{N_e} = 3600;$$

- heat loss to the cooling medium:

$$q_w = \frac{Q_w}{N_e} = \sum \frac{G_w}{N_e} c_w (T_{w2} - T_{w1}),$$

where G_w – hourly consumption of cooling water, kg/h; c_w – heat capacity of water, kJ/(kg×K); T_{w2}, T_{w1} – temperature of the cooling water at the outlet and inlet of the engine jacket space, K;

- heat loss with exhaust gases:

$$q_{hg} = \frac{m_g}{N_e} c_{pg} T_g - \frac{m_a}{N_e} c_{pa} T_0,$$

where m_g, m_a and T_g, T_0 are the masses and temperatures of gases at the outlet of the engine and air at its inlet; s_{pg} and c_{pa} – isobaric heat capacities of exhaust gases and air;

The residual term of the heat balance, which includes unaccounted heat losses:

$$q_{uhl} = \frac{Q_{uhl}}{N_e} = q_f - (q_e + q_w + q_{hg}).$$

Unaccounted heat losses q_{uhl} include radiation heat given off from the heated surfaces of the engine to the environment, chemical underburning of the fuel and its entrainment with the exhaust gases, as well as heat equivalent to the kinetic energy of the exhaust gases.

In two-stroke low-speed engines, a certain part of the energy is spent on organizing purge (Fig. 7.2).

Depending on the operating mode of the engine, the quantitative ratio of the components of the

For preliminary review

thermal balance can change significantly, which is clearly visible in Fig. 7.3, which shows the dependencies for the MAN L28/32 four-stroke engine as a function of its load.

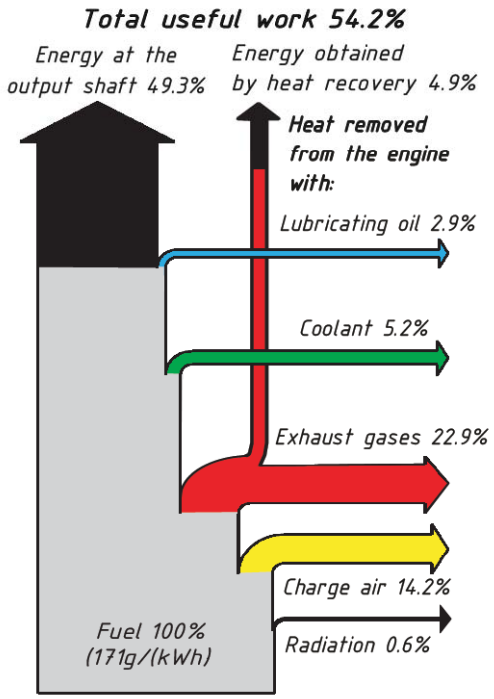


Figure 7.2 – Graphic display of heat and power flows in a low-speed diesel engine 12K98ME-S with a power of 68640 kW at 94 min^{-1} with an exhaust gas heat recovery system (adapted from [1])

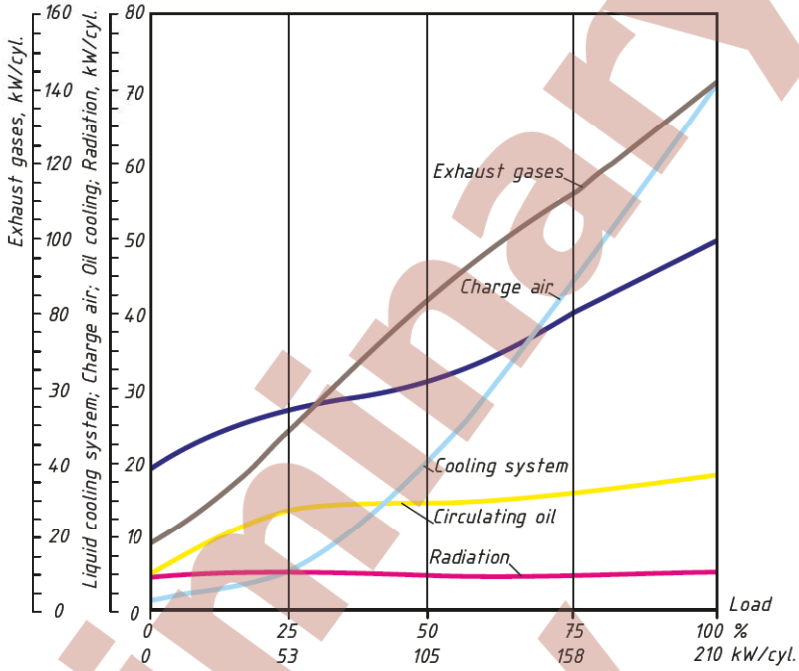


Figure 7.3 – Change in the components of the thermal balance of the MAN L28/32 four-stroke medium-speed diesel engine as a function of its load (adapted from [2])

7.2 Heat exchange between the engine working space and the cooling medium

As can be seen from the engine heat balance equation (7.1), part of the heat obtained during fuel combustion is given to the walls that limit the working space of the engine – the bushing, the fire bottom of the cover and the fire bottom of the piston.

If we consider the heat exchange process in the cylinder as steady, then the amount of heat transferred to the walls can be expressed by the relationship:

$$Q_w = \alpha_i (T_g - T_{cw}) A_\Sigma ,$$

and the heat flux creating the thermal stress of the cylinder walls will be determined as the ratio of the amount of heat transferred from the gas to the walls to the heat exchange surface area, kW/m^2:

$$q_w = \frac{Q_w}{A_\Sigma} = \alpha_i (T_g - T_{cw}),$$

where A_Σ – heat exchange surface area; T_g, T_{cw} – temperatures of gases in the cylinder and walls; α_i – heat transfer coefficient.

Heat exchange between the walls of the engine working space and the gases occurs throughout the entire cycle, but it reaches its greatest intensity during the combustion process and part of the expansion process, when heat release is most intense. As a result, the greatest value of heat flows falls on the fire bottoms of the cover and piston, since in this part of the cycle their surface makes up the majority of the heat exchange surface. This same area accounts for the greatest temperature difference between the gases and the walls, as a result of which about half of all heat losses occur during this period. The intensity of heat transfer is mainly determined by the dynamics of heat release, and the main sources are the convective component of the total heat transfer, as well as radia-

For preliminary review

tion from burning atomization plumes, the flame temperature in which (2300...2500 K) is much higher than the temperature of the gases in the cylinder. Therefore, to ensure the operability of the elements of the cylinder-piston group, the temperatures of the sleeve and piston crown in the area of the first compression ring are taken as limiting values.

In other parts of the cycle, the intensity of heat removal is much lower, and, in some cases, it can be of an alternating nature.

During the filling and compression processes, predominantly convection heat transfer occurs. If the wall temperature T_{cw} is greater than the temperature of the working fluid T_g, then heat from the walls is transferred to the gas, which occurs during the filling process and at the beginning of the compression process. Compression and heating of the working fluid from the walls leads to an increase in its temperature, which at some point exceeds the temperature of the walls and the heat flows change their direction. In total, during the filling and compression processes, 5...8% of the heat from the total losses into the cooling medium is transferred to the walls.

During the expansion process, the intensity of heat transfer decreases due to a decrease in gas pressure and temperature. However, due to the increase in the heat-receiving surface of the bushing, heat losses remain quite high and amount to 30...40% of the total heat losses.

During the release process, heat from the gases is transferred to the walls mainly by convection and accounts for 5...15% of the total losses into the cooling medium.

Heat transfer in engines. As heat is supplied from gases to the walls of the cylinder, their temperature increases. At the same time, the temperature pressure and the amount of heat removed from them increases. At some point, thermal equilibrium is established. The temperature values corresponding to this state in detail are called stationary temperatures.

Due to the complexity of the nature of heat transfer caused by various types of heat transfer, a certain simplified scheme is used in practical calculations of cooling systems.

To a first approximation, the heat transfer process in an engine cylinder can be divided into three successive stages (Fig. 7.4):

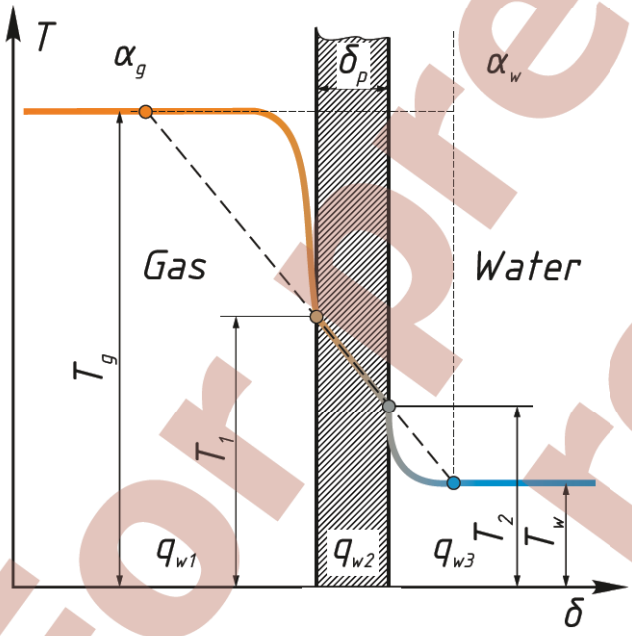


Figure 7.4 – Scheme of heat supply and removal for a flat wall: T_1, T_2 – temperatures of a conventional wall of thickness δ_p on the side of the combustion chamber and on the cooling side, α_g, α_w – heat transfer coefficients from gases to the wall and from the wall to water

- heat transfer from hot gases with temperature T_g to the walls (bushing, piston bottom, cover), which are conventionally taken as a flat wall with a thickness of δ_p, m (one main layer of the wall is considered without taking into account carbon deposits and deposits);

- heat transfer due to thermal conductivity through the wall with the coefficient of thermal conductivity through the main layer – λ_p, kW/(m^2×K);

- heat transfer from the walls of the CPG to the coolant with temperature T_w.

Having accepted the heat exchange process is assumed to be stationary, i.e. when the temperature field does not depend on time and changes only along the coordinates of the space in which heat exchange occurs, we obtain for the three given heat transfer processes in the cylinder the val-

For preliminary review

ues of heat flows, kW/m^2:

$$q_{w1} = \alpha_g \left(T_g - T_1\right);$$

$$q_{w2} = \frac{\lambda_p}{\delta_p}\left(T_1 - T_2\right);$$

$$q_{w3} = \alpha_w \left(T_2 - T_w\right);$$

where α_g, α_w are the heat transfer coefficients of gas and cooling water; T_1, T_2, T_w – wall temperatures, respectively, on the gas side, on the cooling side and on the cooling water side.

The ratio λ_p/δ_p is determined as:

$$\frac{\lambda_p}{\delta_p} = \sum_{i=1}^{n} \frac{\lambda_i}{\delta_i},$$

where i is the number of layers of a conditional wall – soot, main layer (metal), scale; λ_i, δ_i – heat transfer coefficients and thickness of these layers.

Total temperature difference during heat transfer from gases to coolant (water or oil):

$$T_g - T_w = q_{w\Sigma}\left(\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}\right),$$

where is the total heat flow:

$$q_{w\Sigma} = \frac{T_g - T_w}{\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}} = k\left(T_g - T_w\right),$$

where $1/\alpha_g$ is the thermal resistance to heat transfer from the gas to the wall; δ_p/λ_p – thermal resistance to thermal conductivity through the wall; $1/\alpha_w$ – thermal resistance to heat transfer from the wall to the coolant.

The overall heat transfer coefficient from gases to the coolant in general terms, kW/(m^2×K):

$$k = \frac{1}{\frac{1}{\alpha_g} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_w}}.$$

In a stationary heat exchange process, temperatures T_1 and T_2 are taken as steady. In the actual process, on the surface layer of the main wall, due to the variable nature of the heat supply and loss in the soot layer, the temperature is somewhat different from that accepted as the calculated T_1, including due to temperature fluctuations. These fluctuations (3...5% of T_1) along the thickness δ_p are damped and the temperature at a certain thickness (1...1.5 mm from the interface) practically stabilizes. This is explained by the high thermal inertia of the metal, due to which the temperature of the parts does not follow the change in the temperature of the gases in the cylinder. Small temperature fluctuations are observed only on the very surface of the walls of the combustion chamber.

7.3 Temperature fields of parts of the cylinder-piston group

The main indicators of the thermal state of parts of the cylinder-piston group are temperatures in the most critical places and in places where they interface with other parts (Fig. 7.5). The permissible temperature level at each point depends on the operating conditions of the parts. So, for a friction pair: «ring-cylinder bushing», the maximum permissible temperature is determined by the lubrication conditions. Where there is no friction pair, higher temperatures may be acceptable. The thermal state of structural elements, along with other factors, such as mechanical stresses from gas pressure, tightening forces of anchor ties and fastening studs, significantly affect the reliability of the engine and the service life of its main elements.

For preliminary review

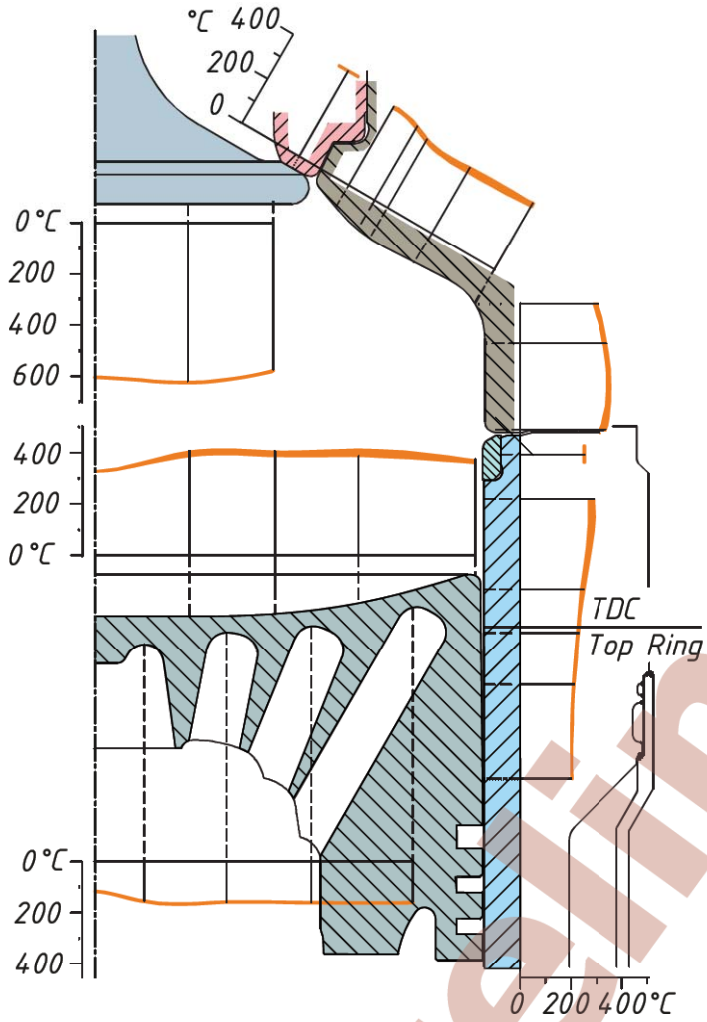


Figure 7.5 – Temperatures on the surfaces of the elements of the cylinder-piston group of the RT-flex 82C two-stroke crosshead engine at a rated power of 4520 kW/cyl., at 102 min^{-1} (adapted from [3])

Both thermal and general stress states are clearly presented in the form of temperature fields and stress fields, which show not only the maximum temperature values on the surface of parts, but also the temperature distribution inside them. Typically, such a distribution is presented in the form of isotherms, lines connecting points with the same temperatures, or in the form of zones with a given temperature gradient, highlighted in one color (Fig. 7.6).

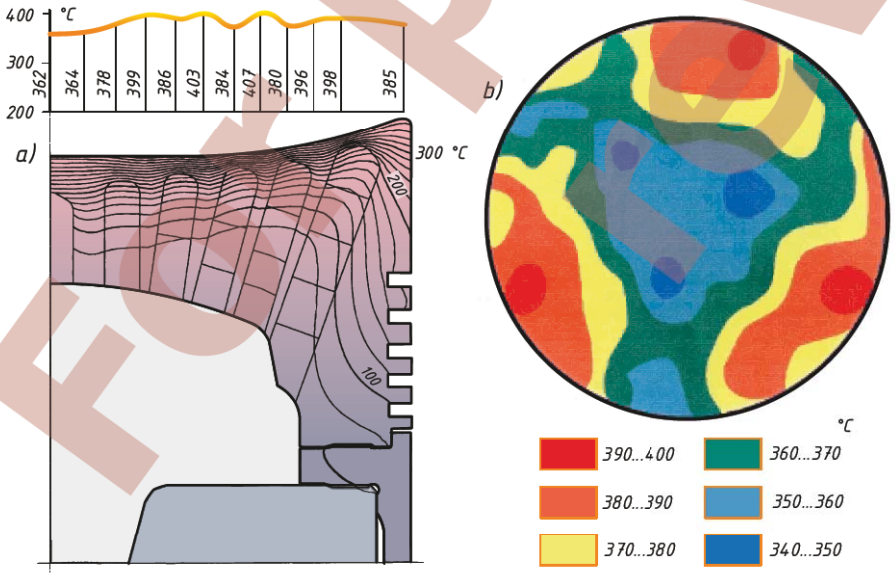


Figure 7.6 – Calculated temperature distribution in the piston head of the RTA96C engine at rated power mode (*a*) and the piston surface temperature measured on the 11RTA96C engine (53300 kW at 94 min^{-1}) (*b*) (adapted from [4])

Assessing the thermal state of parts is a rather complex task, an effective solution of which can only be achieved using three-dimensional digital modeling methods using computer technology.

For preliminary review

When solving such problems, the finite element method is usually used, into which the housing under consideration is divided (usually elements of a triangular or rectangular cross-section). It allows you to determine temperature fields in parts of any shape with an accuracy depending on the accuracy of setting the boundary conditions.

Temperatures at which the strength characteristics of the material, its chemical structure or the properties of other parts in contact with it are not deteriorated are accepted as acceptable. Permissible temperature values depend on the material of the part, manufacturing method, design features, operating conditions, etc.

When designing for the most critical elements, the temperature fields inside the parts are determined (Fig. 7.5), the influence of various operational factors on the thermal state of the part is analyzed, for example, carbon deposits on the gas side, contamination on the cooling side, cooling water temperature, etc. Using these fields, the most thermally loaded places are identified and measures to reduce temperatures in them are determined (change in shape, replacement of material).

7.4 Thermal stresses of parts of the cylinder-piston group and factors limiting the temperature of the working surfaces

The heat flows passing through the material of the parts of the cylinder-piston group create thermal and mechanical stress in them – thermal stresses σ_t are proportional to the temperature difference ΔT across the wall thickness δ_p.

Exceeding permissible levels of thermal stress in parts of the cylinder-piston group can have a number of negative consequences, which include:

- the surfaces of the combustion chamber walls reach temperatures at which plastic deformation of the metal is possible. Combined with high pressure in the cylinder, this can lead to destruction of parts or burnout of metal in the hottest places;
- high temperature gradients (large differences between the temperature at the fire surface and the temperature on the cooling side) can lead to unacceptable thermal stresses. The hotter layers of metal near the fire surface tend to expand, which is prevented by the underlying colder layers that create compressive forces. At the same time, the metal layers on the cooling side are loaded with tensile forces. Exceeding the tensile strength of the metal under thermal stresses leads to cracking of the wall material and failure of structural elements;
- in highly thermally loaded engines, frequent changes in the temperature conditions of the elements of the cylinder-piston group lead to the accumulation of thermal fatigue in them;
- at elevated temperatures on the surface of the bushing in the area of action of the piston rings, the strength of the oil film is disrupted, the oil loses its lubricating properties, flows down, oxidizes, and forms varnish deposits. As a result, the coefficient of friction increases. This leads to an even greater increase in temperature, wear of the piston rings and bushings, scuffing of the pistons and engine failure;
- uneven heating of the cylinder liner along the height and circumference leads to its deformation and violation of the geometry. Similar phenomena occur when the piston is heated unevenly. This leads to a reduction in some places of the gaps between the piston and the bushing, and, as a consequence, to disruption of lubrication, emergency wear and scuffing.
- when the pistons are oil cooled, the increased surface temperature on the cooling side can lead to intense coke formation, disruption of the cooling mode, overheating and emergency destruction of the piston.

Modern trends towards increasing the level of engine boost (increasing p_z and p_e) lead to an increase in the mechanical stress of the main elements of the cylinder-piston group and an increase in the density of heat flows $q_{w\Sigma}$ through them. Mechanical stresses decrease with increasing wall thickness δ_p, but at the same time temperature stresses increase. To ensure reliable operation of the engine, it is required that the total mechanical and temperature stresses do not exceed permissible values. When designing engines, this is one of the main criteria when choosing design solutions and materials used.

For preliminary review

Due to the exceptional difficulty of directly measuring thermal stresses in the parts of the cylinder-piston group, an indirect indicator is used to evaluate them – the temperature gradient dt/dx, showing the change in temperature in the parts along the thickness of the metal layer. In modern engines this indicator lies within the range $dt/dx \approx 4…5$ °C/mm.

For the fire bottoms of the working cylinder covers, the temperature is limited by the presence of various stress concentrators (holes for valves, channels, as well as stagnant zones for cooling water) and, depending on the material, has the following values: 170...250°C for light alloys; 250…300°C for cast iron; 300…350°C for steel.

According to experimental data, the temperature values for the piston heads on the firing surface side, depending on the material used, are 300...350°C for light alloys; 400…450°C for cast iron; 450…500°C for steel.

For modern low-speed engines, temperatures at characteristic points of parts of the cylinder-piston group should not exceed the values given in table. 7.1.

Table 7.1 – Temperatures at characteristic points of parts of the cylinder-piston group of modern low-speed engines, °C

Surface and/or its location	Meaning
Piston head	450…550
Piston in the area of the 1st piston ring	160…180
Bushing in the combustion chamber area	180…200
Bushing in the area of the 1st piston ring	140…160
Bottom of cylinder cover	300…400
Exhaust valves	600…700

Thermal stress is the main factor limiting the cylinder power of the engine when it is boosted. Given in table. 7.1 the values of temperatures and temperature gradients are close to the maximum permissible. In contrast to the mechanical strength margin, the thermal state margin of the cylinders is small and quickly decreases when diesel engines are boosted.

The temperature of the working surface of the cylinder liner is mainly limited by the possibility of so-called low-temperature corrosion.

Since marine engines may use fuels with a high sulfur content, when burned they produce sulfur dioxide, SO_2. Through a series of intermediate reactions and interaction with water vapor, it forms a condensate of sulfuric acid H_2SO_4.

The corrosive effect of concentrated sulfuric acid condensate at high temperatures is insignificant, but in a 20...60% solution with water vapor condensed on the surface of the cylinder liners, its activity increases greatly. This phenomenon is typical for engine operating modes at low loads. It is possible to avoid increased corrosive wear of the bushing mirror only if the condensation temperature is lower than the temperature of the bushing mirror throughout the entire piston stroke. The nature of the change in condensation temperatures (dew point temperature) and working surface along the height of the cylinder liner of a low-speed engine is shown in Fig. 7.7. To prevent the bushing surface from reaching the dew point temperature, in modern engines running on heavy fuels, its cooling is carried out only in the upper part.

The permissible temperatures of the internal cavities of oil-cooled pistons and annular grooves depend on the oil used, in particular on its characteristics such as varnish formation temperature. For oils used in the navy, this temperature is in the range of 200...300°C. Usually, when designing engines, they strive to ensure that the temperature of the piston bottom on the cooling side is 40...50 °C below the varnish formation temperature, and in the area of the first sealing ring the piston temperature does not exceed 160 °C.

The exhaust valves have a large surface facing the combustion chamber, which leads to their strong heating. Some of the heat is also transferred to the valve disc and spindle during the release of gases that wash their surfaces at high speed. The highest temperatures are concentrated in the central part of the valve disc, which quickly decrease towards the point of contact of the disc with

For preliminary review

the seat (Fig. 7.8). As shown In Fig. 7.8, the hottest area is the center of the valve end with a temperature of 630°C. The area near the valve seat is much cooler, with a temperature of about 390°C. Strong temperature gradients lead to increased mechanical stress due to thermal stress in this area.

Operation at high temperatures and high engine load reduces the resistance of valve materials to high temperature corrosion and shortens their service life.

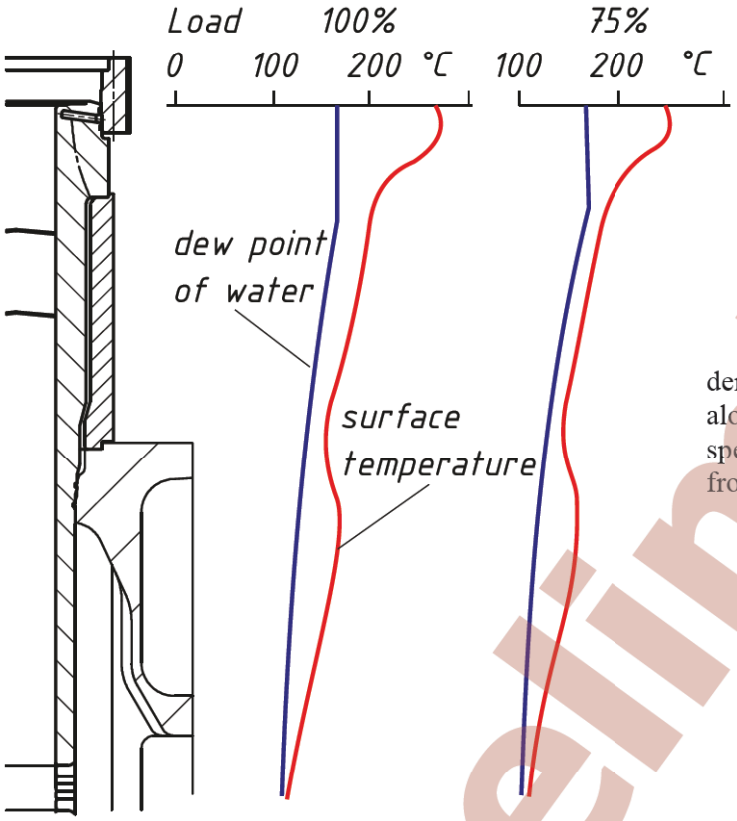


Figure 7.7 – Character of changes in condensation and working surface temperatures along the height of the cylinder liner of a low-speed two-stroke diesel engine (adapted from [5])

The effect of high temperature corrosion in engines operating on heavy fuel oil occurs due to the large amount of vanadium and sodium contained in the heavy fuel oil (up to 600 and 200 min^{-1}, respectively). Vanadium in fuel is contained in the form of fuel -soluble complex organometallic compounds, and sodium is mostly in the form of *NaCl* crystals or an aqueous solution of its salts. During combustion, these elements form various compounds called sodium vanadyl vanadates. In structure, these are semi-liquid salts with a low melting point and high adhesive ability.

The most active compounds of this group are $Na_2O \cdot 6V_2O_5$ and $5Na_2O \cdot 12V_2O_5$. During their solidification on the metal surface in the temperature range of 530...600°C, atomic oxygen is released, which reacts with the metal and forms oxides that diffuse into the overlying layer of deposits. As the deposits remelt, oxygen is absorbed from the exhaust gases. Thus, sodium vanadyl vanadates act as an oxygen carrier from exhaust gases to the metal surface, causing intense corrosion.

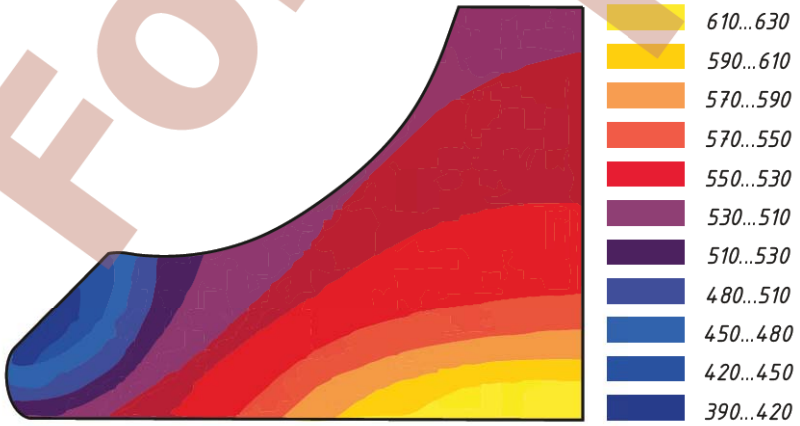


Figure 7.8 – Typical temperature distribution (°C) for the exhaust valve of a modern diesel engine made of Nimonic alloy (adapted from [6])

For preliminary review

As shown in Fig. 7.8, the exhaust valves of modern engines operate in a critical temperature range, which contributes to the occurrence of high-temperature corrosion. To reduce the average temperature, the seat cooling rate is increased and heat is intensively removed from the valve stem guide sleeve.

To reduce the intensity of high-temperature corrosion, special glass-metal coatings are applied to the valves of some modern high-performance diesel engines, which prevent direct interaction of the metal with oxygen.

In two-stroke engines, part of the heat is removed from the rod through the valve tappet drive oil, the valve guide, and the valve is also cyclically cooled by purge air.

The reliability of engine operation also decreases with an increase in the temperature level of other parts – nozzles, exhaust valves, jumpers between purge ports, etc.

As already noted, direct monitoring of the thermally stressed state of engine structural elements is extremely difficult. For this reason, the thermal state of the cylinder-piston group during operation is assessed by indirect indicators, which include:

- position of the fuel rail;
- exhaust gas temperatures per cylinder;
- cooling water and oil temperatures;
- temperature changes in cooling systems;
- air temperature;
- average indicator pressure;
- crankshaft rotation speed, etc.

Such an assessment does not guarantee timely detection of thermal overloads. It requires a comprehensive analysis of engine operating parameters, which presupposes that the maintenance personnel have certain experience and a deep understanding of the relationship between individual aspects of engine operation.

7.5 Purpose and composition of cooling systems

As follows from the engine thermal balance equation (7.1), part of the heat not converted into work in the cylinders, as well as the heat released as a result of friction between parts, must be removed to the environment, maintaining the optimal thermal regime of the engine. To achieve optimal operating process parameters, it is necessary to maintain the supply air temperature within a strictly defined range. For effective lubrication, it is necessary to remove heat from the circulating oil and maintain the temperature of the friction pairs at a strictly defined level. All these functions are assigned to the engine cooling system.

Modern marine engines use fresh water for cooling, which is continuously circulated through a closed system. The heat transferred to the water in the engine cooling circuits is removed to the environment in the central heat exchanger by pumping sea water through it.

The cooling systemsof most modern engines are made of two circuits: high temperature (HT) and low temperature (LT) (Fig. 7.9).

The HT circuit cools the cylinders, cylinder heads and turbochargers. The high temperature levels characteristic of the HT circuit make it possible to minimize temperature differences in the elements of the cylinder-piston group, which reduces their thermal stress and prevents low-temperature corrosion. A separate pump supplies water to the distribution channel, usually located in the cylinder block. Next, the water enters the water jackets of the cylinders and then, through the connecting parts, into the cylinder covers. From the covers, water flows through a pipe into the collection manifold and then to the thermostat, which automatically maintains the temperature in the circuit at a given level. If the water temperature is higher than the required value, the thermostat will allow most of it to flow into the central cooler, and less to the pump inlet. Thus, in the thermostat there is a constant redistribution of water flows, by mixing which the desired temperature is obtained. In parallel with the cylinders, part of the water flows to the turbo compressors (if such cooling is provided for by the design of the turbocharger).

For preliminary review

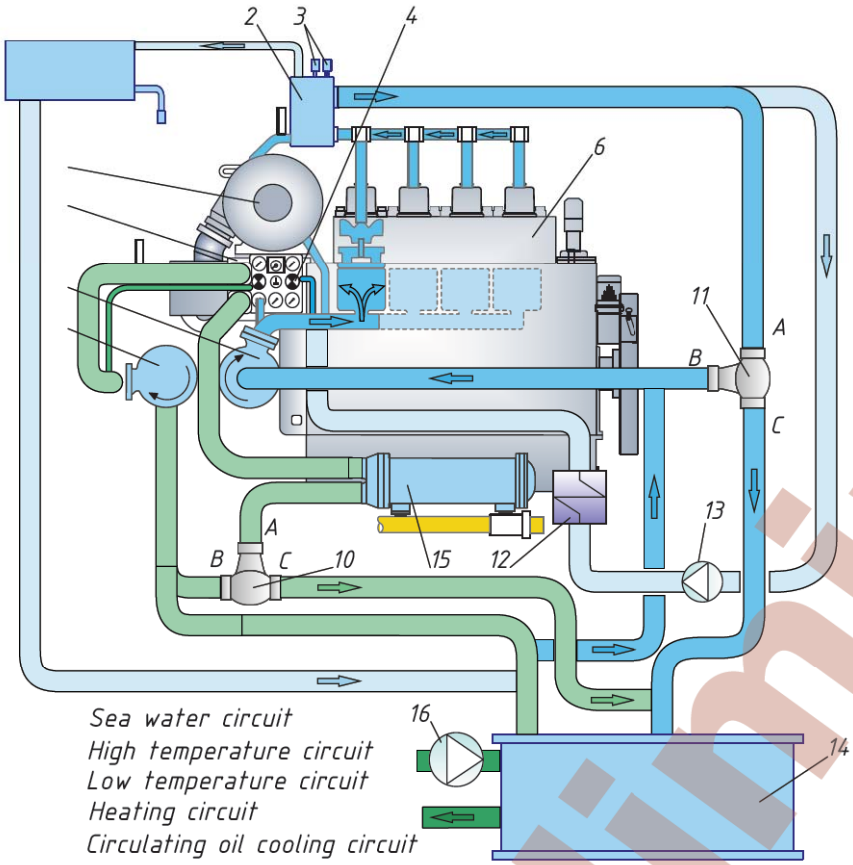


Figure 7.9 – Diagram of the cooling system of a four-stroke marine engine: 1 – expansion tank; 2 – ventilation box; 3 – alarm sensor; 4 – pressure gauge; 5 – turbocharger; 6 – engine; 7 – charge air cooler; 8 – water pump of the low-temperature circuit; 9 – water pump of the high-temperature circuit; 10 – low-temperature circuit temperature control valve; 11 – valve for temperature control of the high-temperature circuit; 12 – heater; 13 – pump; 14 – central cooler 15 – lubricating oil cooler; 16 – sea water pump (adapted from [7])

The LT circuit includes charge air and lubricating oil coolers, through which water is pumped by a pump identical to the HT circuit pump. The water temperature in the circuit is controlled by a thermostat.

To remove air from the system, after cooling the working cylinders and turbocharger, water enters a separation box (deaerator) which is connected to the expansion tank. The pumps of both circuits take water from the expansion tank, which, located above the engine, creates a static pressure of 0.07...0.15 MPa in the system, necessary for the efficient operation of the pumps. To heat water in front of the engine (for example, when starting from a cold state), an additional heater and a pump with an independent drive, usually electric, are connected to the LT circuit. Before starting the engine, the LT circuit is warmed up. Check valves in the circuits ensure the movement of water in specified directions.

Piston cooling in all modern engines is carried out by oil from the circulation system, which is cooled by water from a low-temperature circuit.

Water with special additives (anti-corrosion, anti-cavitation) are used as coolants in the internal closed circuit of marine diesel engines.

The main physical properties of fresh water and oil used in cooling systemsare given in table. 7.2.

Table 7.2 – Main characteristics of coolants

Characteristic	Liquid	
	Water	Oil
Density, ρ at 80°C and 0.1 MPa, kg/m^3	971.4	856.0
Specific heat capacity, s_r, kJ/(kg×K)	4.196	2.089
Kinematic viscosity ν, ×10^{-6} m^3/s	0.365	36.7

Heat exchange in the inner space of the engine has a significant impact both on the flow of the operating process in the working cylinders and on the thermal stress of the cooled elements of the engine structure. The main factors that determine the mode of heat transfer from the walls to the coolant are: the temperatures of the wall and the coolant, the pressure in the cooling cavity, the

For preliminary review

mode of fluid flow and its physical and chemical characteristics, the shape and cross-section of the flow part of the cooling cavity.

Heat removal from the fresh water of the internal circuit is ensured by a central cooler, which is pumped with sea water by a special pump.

The cooling intensity is characterized by the temperatures of the cooling water at the inlet and outlet of the corresponding circuit, as well as the magnitude of their difference (Table 7.3).

Table 7.3 – Temperature limits of cooling media in cooling systems, °C

	Low-speed	Medium speed	High speed
Cooling water of internal circuits:			
- engine inlet	65…75	70…82	76…87
- exit from the engine	75…80	80…90	80…110
- difference	5…10	5…10	4…8
Preheating	50	40…50	40…45
Heating when operating on heavy fuel	60…70	60…70	–
Sea water:			
- entrance to the cooler, max.	32…38	32…38	32…40
- exit from the cooler, max.	≤ 50	≤ 50	≤ 50

Typically, the temperature of water in cooling systemsis limited by its boiling point at atmospheric pressure, but recently, in order to reduce the thermal stress of parts, closed cooling systemsin which the coolant is under pressure have been used on marine engines. The operating temperature in such circuits is usually higher than the boiling point of water at atmospheric pressure and can reach up to 120°C. Such systemsor their individual circuits are used on all types of modern engines, including low-speed ones.

In general, the cooling system removes up to 40% of the total heat of fuel combustion into the seawater. The share of heat rejected into water and oil for various types of marine engines is given in table. 7.4.

Table 7.4 – Fraction of heat transferred to water and oil

Engine's type	Cooling water	Cooling oil
Low-speed	14…20	6…15
Medium speed	12…25	10…15
High speed	30…50	5…15

7.5.1 High temperature circuit

In Fig. 7.10 and 7.11 show how the elements of the cylinder-piston group of four- and two-stroke marine engines are cooled with water from a high-temperature circuit.

Water is pumped into an annular cavity around the top of the cylinder liner, called the cooling jacket. In highly accelerated engines, both two- and four-stroke, in order to maintain mechanical strength at acceptable levels of thermal stress, water from the jacket is supplied to holes made in the upper flange of the bushing, which are as close as possible to its firing surface (Fig. 7.10 *a*, 7.11 *a*). Next, through the holes or transition manifolds (Fig. 7.10 *a*) (transition tubes, Fig. 7.11 *c*) cooling water flows to the cylinder covers. For four-stroke engines with two-tier cooling of the cylinder cover (Fig. 7.10 *a*), water is directed by a separating plate into the cavity of the first tier, where it moves through drillings along the fire bottom. This makes it possible to reduce the thermal stress of the fire bottom without reducing its mechanical strength. Part of the water is discharged into the annular cooling grooves of the exhaust valve seats (Fig. 7.10 *b*). Next, the water enters the central cavity and, cooling the nozzle, passes into the cavity of the second tier (Fig. 7.10 *a*), from where the drainage collector is discharged.

For preliminary review

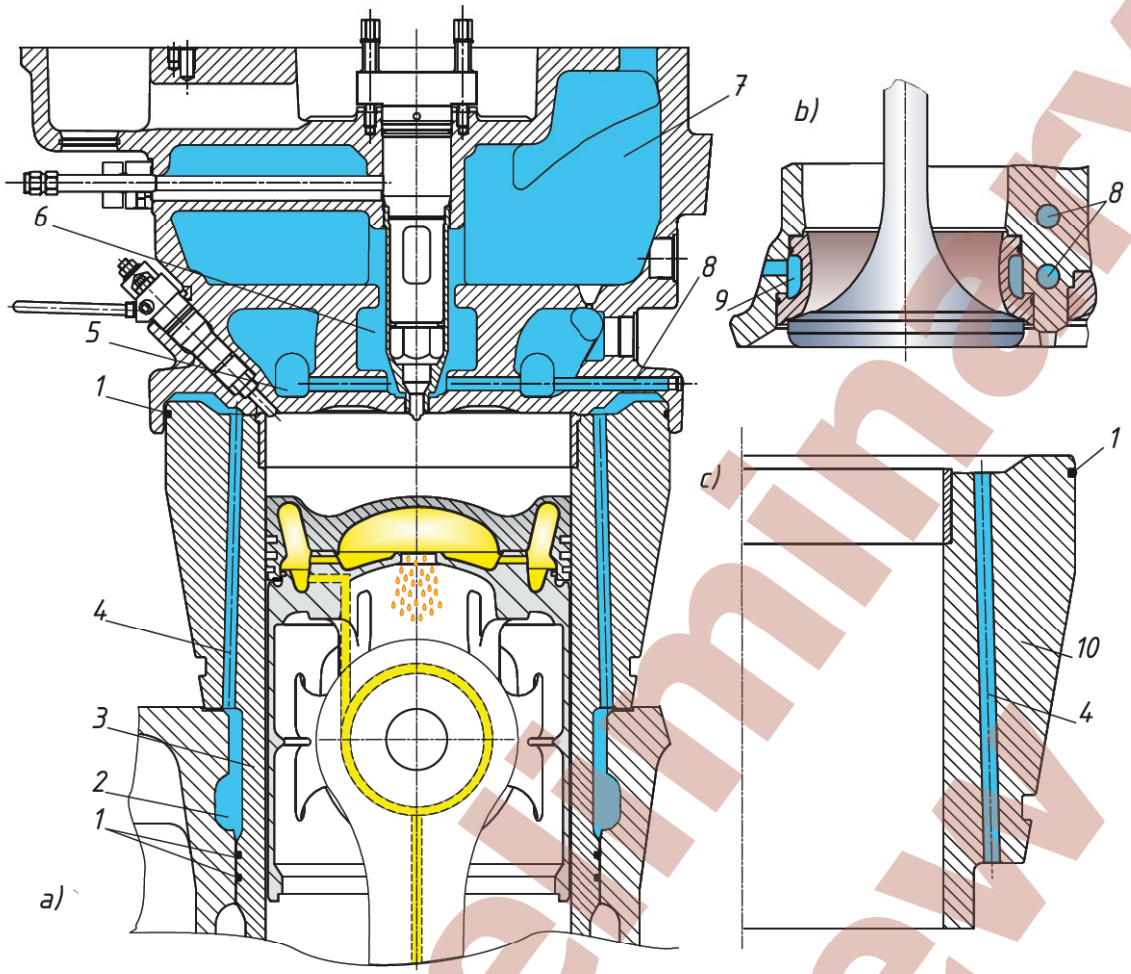


Figure 7.10 – Cooling of the main elements of a four-stroke medium-speed engine 46 series from Wärtsilä. *a* – cooling of the elements forming the engine working space; *b* – cooling of exhaust valve seats; *c* – channel cooling of the firing surface of the cylinder liner; 1 – sealing rings; 2 – cooling jacket; 3 – cylinder liner; 4 – channels for cooling the firing surface of the cylinder liner; 5 – annular water manifold for supplying water to the fire bottom of the cylinder cover; 6 – fuel injector cooling cavity; 7 – cooling cavity of the second tier; 8 – radial drillings for supplying cooling water to the fire bottom of the cylinder cover; 9 – cavity for supplying water for cooling the exhaust valve seat; 10 – upper collar of the cylinder liner (adapted from [9])

In two-stroke engines, water from oblique-radial drillings along the fire bottom of the cylinder cover enters the annular manifold of the formations between the cover and the exhaust valve assembly (Fig. 7.11 *b*). From here, part of the water is supplied to cool the valve seat, and the remaining flow is supplied to cool the valve assembly.

In some cases, on low-speed engines, sea water is used to cool the charge air, which is supplied to the cooler by a separate pump.

For high-performance, medium-speed engines, the charge air cooler is divided into high-temperature and low-temperature sections to achieve maximum heat recovery. By adjusting the intensity of water flows in both sections, stable temperatures of the charge air at the engine inlet are achieved, which is especially important at low and medium loads, when the air needs to be heated.

7.5.2 Piston cooling

To cool the pistons in modern engines, circulating oil is used, which greatly simplifies the system for supplying the cooling medium to the moving pistons and, most importantly, minimizes the consequences of leaks in the event of damage to the seals.

Low-speed MAN engines use circulation cooling of the pistons (Fig. 7.12 *a*), in which oil enters

For preliminary review

the piston cavity through a special pipe passing inside the piston rod. The pipe is installed with a gap in the internal drilling of the rod and divides it into two channels. The oil flows through the pipe itself to the piston, and through the annular channel between the pipe and the drilling in the rod, it drains back into the crankcase. Initially, the oil enters the cavity under the fire bottom of the piston and, moving along it, removes excess heat. Next, through drillings in the support ring, the oil enters the annular cavity in the area of operation of the piston rings, ensuring heat removal from it, after which it passes into the cavity serving to cool the piston guide. From this cavity, the oil is drained.

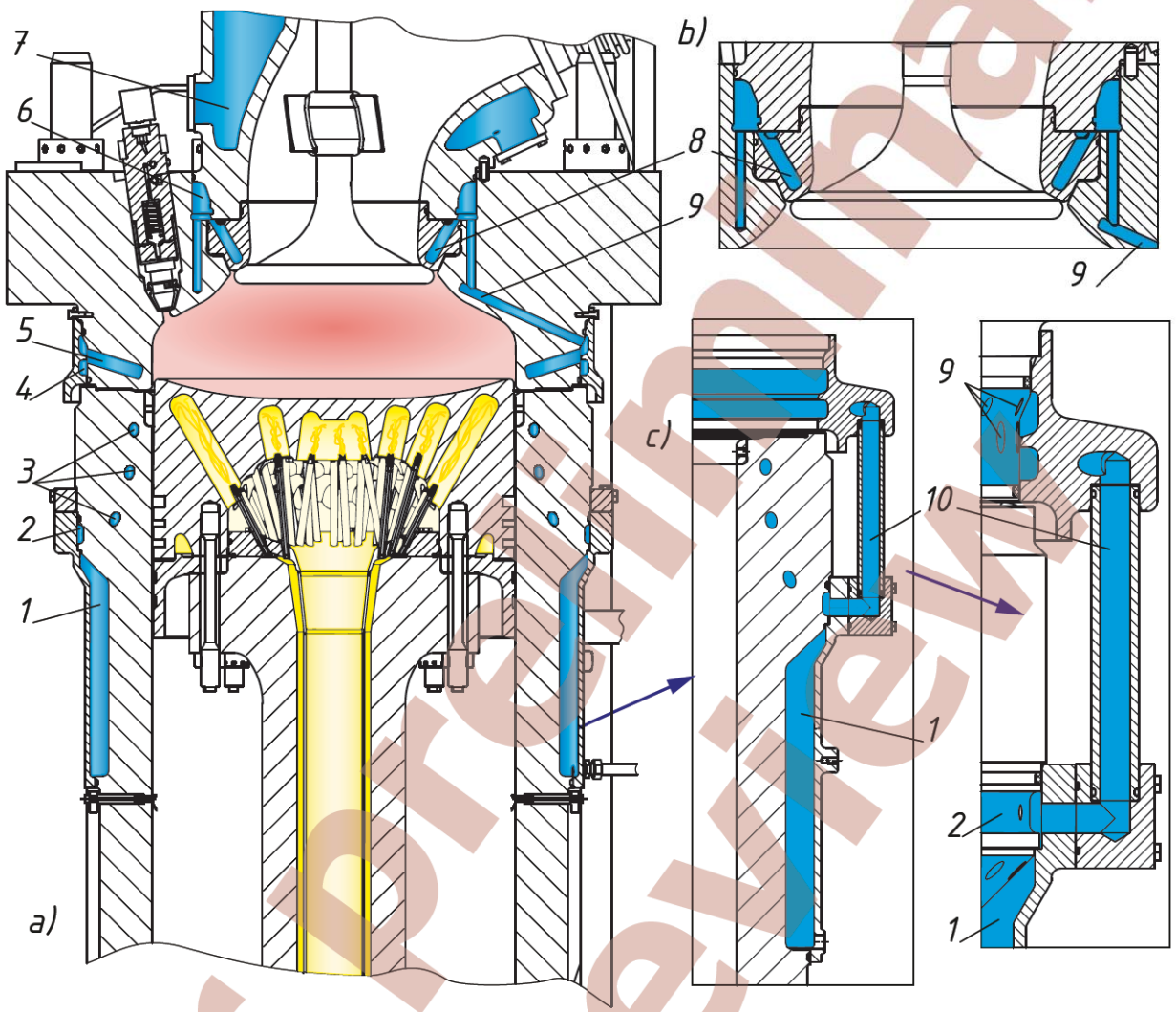


Figure 7.11 – Cooling of the main elements of the two-stroke low-speed engine X-82B from WinGD: *a* – general arrangement of cooling cavities; *b* – exhaust valve cooling; *c* – water supply to the cooling jacket, the firing surface of the bushing and the cylinder cover; 1 – cylinder liner cooling jacket; 2 – annular manifold for supplying water to the drillings in the upper flange of the cylinder liner; 3 – drilling in the upper flange of the cylinder liner to supply coolant to the firing surface; 4 – ring manifold for supplying water to radial drillings in the cylinder cover; 5 – radial drillings in the cylinder cover; 6 – ring manifold for supplying water to the radial drillings of the valve seat; 7 – cooling cavity of the valve assembly; 8 – inclined drillings in the valve seat; 9 – inclined-radial drillings in the cylinder cover along the fire surface; 10 – tubes for supplying cooling water to the cylinder cover (adapted from [10])

For modern high-performance four-stroke engines, the most common method of oil cooling of pistons is the so-called circulation-shaker cooling or shaking cooling (Fig. 7.12 *b*, *d*). With this method of heat removal, oil is supplied through drilling in the connecting rod and oil distribution channels in the piston into the annular cavity of the piston head in the area of the piston rings through the throttle insert (Fig. 7.12 *b*). Having filled this cavity, oil under pressure flows into the so-called shaker cavity through radial drillings in the retaining ring. This cavity is formed by the

For preliminary review

bottom of the piston and a special diaphragm made in its guide part. There is a hole in the center of the diaphragm to allow oil to drain back into the crankcase. The cross-sections of the inlet and outlet channels are selected in such a way that the shaker cavity is only partially filled. Under the influence of inertial forces, the oil is alternately thrown to the top or bottom of the cooling cavity, intensively washing the heated areas of the piston (Fig. 7.12 *c*). The heated oil is replaced by incoming new portions, and the used oil flows into the engine crankcase. A high coefficient of heat transfer to the oil is ensured due to the high average speed of the piston and the participation in heat transfer of the entire mass of oil, and not just the boundary layer.

Another option for organizing cooling using a circulation-shaker scheme is shown in Fig. 7.12 *d*. Oil enters the central cavity between the piston bottom and the diaphragm through a special spring-loaded fitting that is in constant contact with the upper head of the connecting rod. Oil flows to the fitting through a drilling in the connecting rod and a system of oil distribution grooves. From the fitting, the oil immediately falls on the inner surface of the fire bottom, cooling it. After this, it flows to the bottom of the cavity and then participates in cooling by shaking. As the piston moves to BDC, under the influence of inertial forces, the oil is pressed against the piston bottom. Some of it is forced out through drillings in the retaining ring into the cooling cavity of the piston ring zone, from where it drains into the crankcase. Thus, both cavities under the firing bottom of the piston act as shaker cavities.

Due to its high efficiency, this cooling method has become widespread on both medium- and low-speed engines. However, in high-performance low-speed engines it is often used in combination with jet cooling. An example of this method of heat removal is shown in Fig. 7.13.

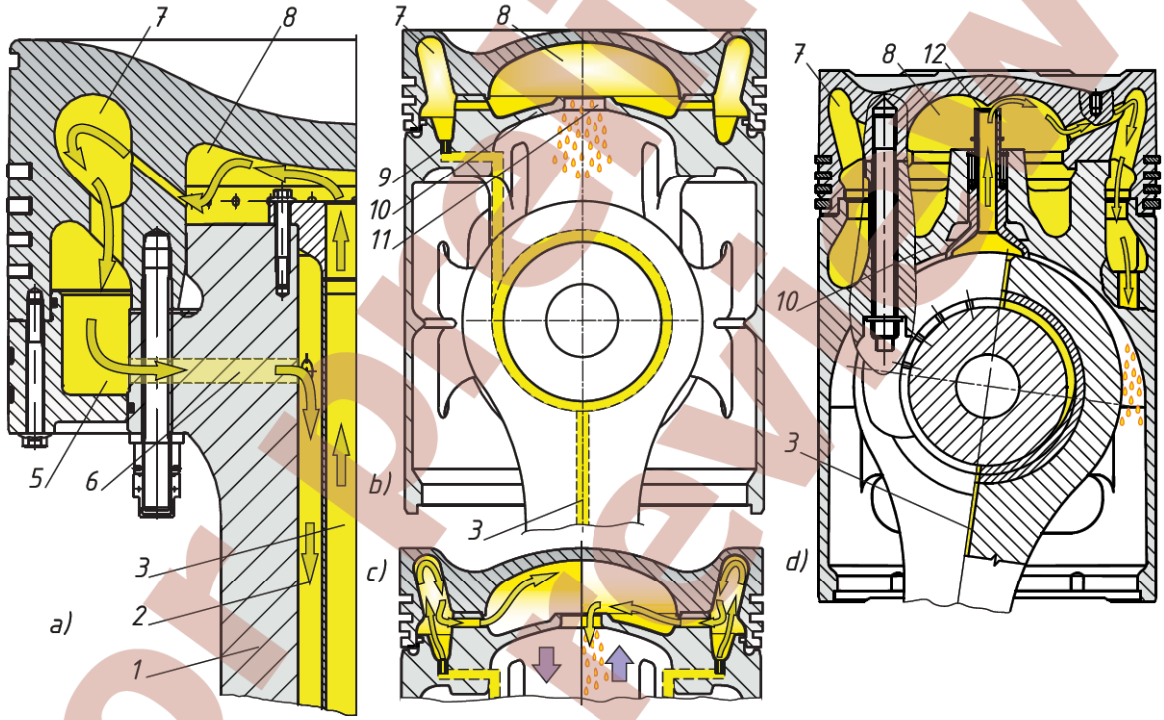


Figure 7.12 – Cooling of pistons of marine engines: *a* – circulation cooling of pistons of low-speed two-stroke engines of the MC-C series from MAN; *b* – circulation-shaker cooling of the pistons of the 46 series engines from Wärtsilä; *c* – scheme of oil flows during circulation-shaker cooling; *d* – circulation-shaker cooling of the pistons of L 32/40 engines from MAN; 1 – piston rod; 2 – drain cooling oil; 3 – channel for supplying oil for cooling the piston head; 5 – cooling cavity of the piston skirt; 6 – oil drain channel; 7 – cooling cavity of the piston ring zone; 8 – cooling cavity of the fire bottom; 9 – throttle insert; 10 – diaphragm; 11 – drain hole; 12 – oil supply fitting (adapted from [9, 11, 12])

A separating diaphragm with nozzle sprayers attached to it (7.13 *c*) is installed on the piston rod flange, each of which is inserted into inclined drillings in the piston head (7.13 *b*). Oil from the circulating lubrication system under pressure enters under the diaphragm, through a channel formed by drilling in the rod and a drain pipe inserted into it.

For preliminary review

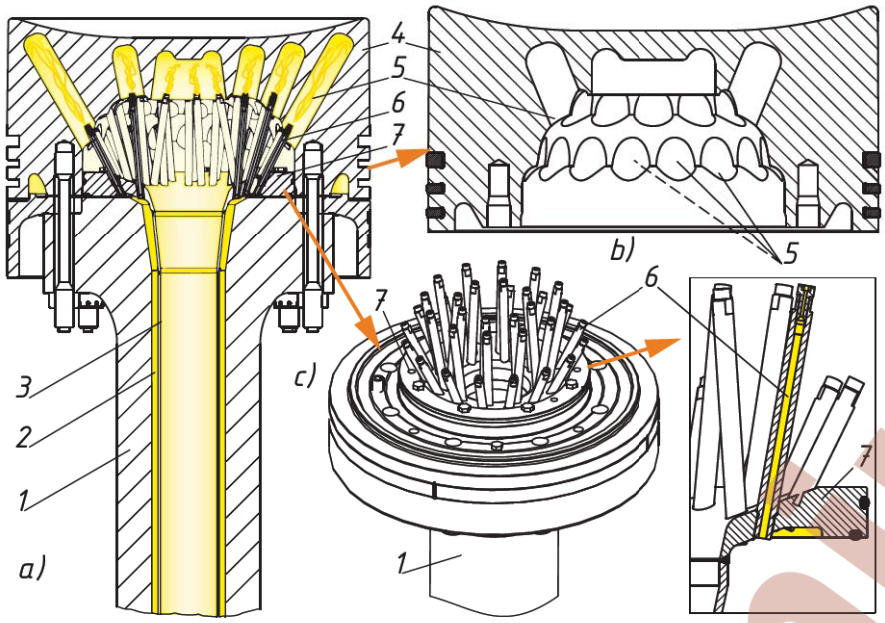


Figure 7.13 – Jet cooling of the piston of a two-stroke low-speed engine X-82B from WinGD: 1 – piston rod; 2 – space between the rod and drain pipe used to supply oil for cooling the piston; 3 – pipe for draining oil cooling the piston; 4 – piston head; 5 – blind drillings for cooling the fire bottom; 6 – oil nozzles for cooling the fire bottom of the piston; 7 – diaphragm (adapted from [10])

When the piston moves upward, when the oil in the piston cavity is pressed downward by inertial forces, jets of oil shoot out from the nozzles, intensively cooling the head in the immediate vicinity of the firing surface. When the piston moves back down, the oil, under the influence of inertial forces, is pressed against the piston bottom, cooling it intensively. During the subsequent upward stroke, the heated oil flows down the drain pipe into the engine crankcase. Each drain point is equipped with a temperature sensor, with which you can control the thermal conditions of a particular cylinder.

A type of jet cooling can be considered a method of removing heat from the piston, which has become widespread in high-speed engines with a relatively small cylinder diameter (Fig. 7.14). Sometimes this cooling method is called fountain cooling.

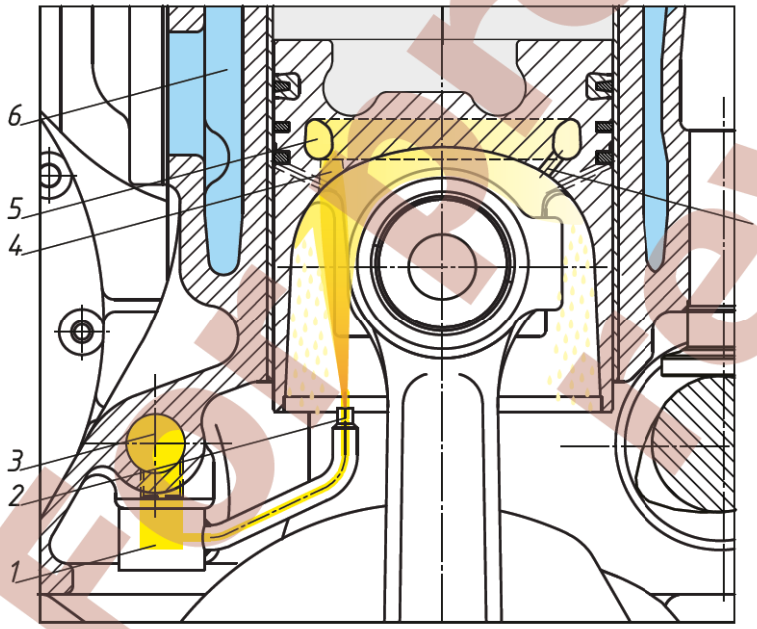


Figure 7.14 – Jet cooling of the piston of a high-speed engine: 1 – oil nozzle housing; 2 – nozzle; 3 – oil line; 4 – receiving cone; 5 – toroidal chamber in the piston; 6 – cooling jacket; 7 – drain hole (adapted from [13])

A special oil nozzle, fixedly installed under the cylinder, is connected to the oil supply channel, which flows to it intermittently when the piston is at BDC. A jet of oil hits the piston bottom and partially enters the toroidal channel in its head through a special conical bore, and partially onto the inner surface of the fire bottom. The oil in the toroidal channel is agitated under the influence of inertial forces, removing heat from the piston ring area, after which it flows into the engine crankcase through special inclined drillings. Oil falling directly on the inner surface of the fire bottom

Preliminary preview

cools it. The oil supply line to the injectors is separated from the main circulation system by a pressure reducing valve configured in such a way that the oil supply begins only when the pressure in the system rises above a certain value necessary to ensure reliable lubrication of the bearings. As a flow interrupter, a system of drillings and grooves is usually used in the camshaft journals and their bearings, which act as a rotating spool pair.

7.6 Charge air cooling

In modern highly accelerated engines, the charge air cooling system plays a key role in organizing the engine's operating process. Despite the name, in modern engines, the function of the system is not only to cool the charge air when the engine is operating at loads close to maximum, but also to heat it when the engine is operating at low and medium loads. For this purpose, engines are equipped with a two-section cooler, one section of which is included in the HT circuit, and the other in the LT circuit (Fig. 7.15). Thus, the air is first cooled (or heated) in the HT circuit section, and then enters the LT circuit section in which the coolant temperature is regulated by a special thermostatic valve. The valve either connects the LT section to the low temperature circuit if air cooling is required, or disconnects the section if there is no need for cooling. The joint operation of the two sections makes it possible to obtain stable air temperatures at the engine inlet, regardless of its operating mode.

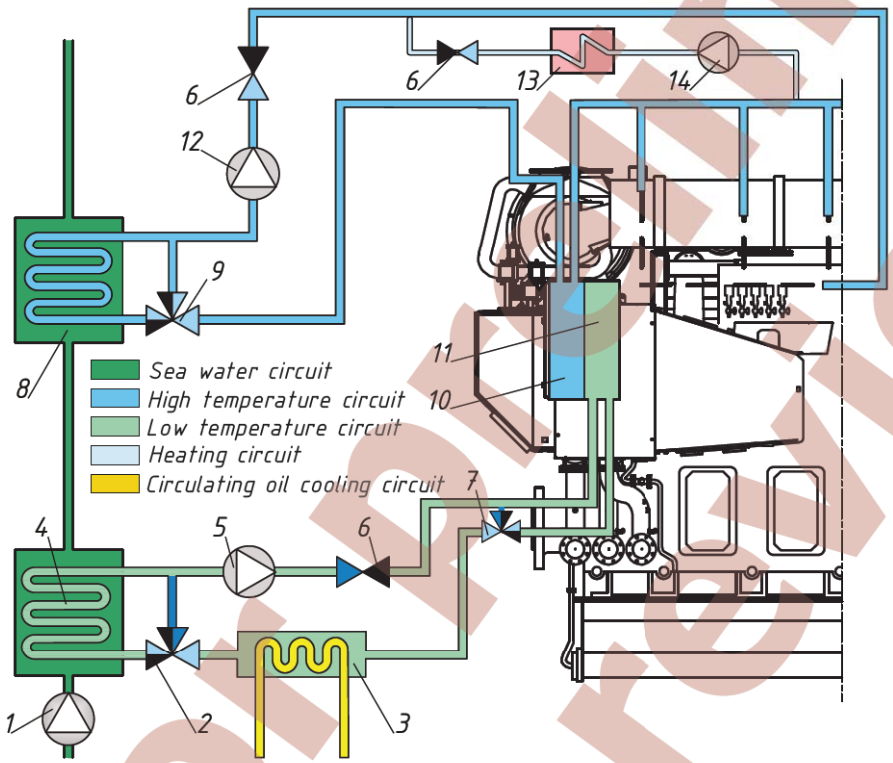


Figure 7.15 – Diagram of connecting the air cooler to the HT and LT circuits of a medium-speed four-stroke engine: 1 – sea water pump; 2 – thermostatic valve LT; 3 – oil cooler; 4 – LT circuit cooler; 5 – LT circuit pump; 6 – shut-off valve; 7 – thermostat of the LT stage of the air cooler; 8 – HT circuit cooler; 9 – thermostatic valve HT; 10 – section LT of the air cooler; 11 – LT section of the air cooler; 12 – – HT circuit pump; 13 – heater; 14 – heating circuit pump (adapted from [14])

Some engines use a special thermostatic valve to control the air temperature at the cooler inlet, allowing water from the high-temperature circuit to pass through the low-temperature section of the air cooler. The valve is controlled by charge air pressure, which directly depends on the operating mode of the engine. An example of such a valve used on engines of the TBD 645 series from DEUTZ MWM is shown in Fig. 7.16.

7.7 Main units and elements of cooling systems

Pumps. Centrifugal pumps are used for the cooling system of diesel engines. They have a fairly high efficiency and are simple in design. The use of centrifugal pumps allows for continuous flow, without pulsation, which ensures uniform cooling of the engine. To protect against the aggressive

For preliminary review

effects of cooling media, especially sea water, pump parts are made of stainless alloys or covered with stainless lining.

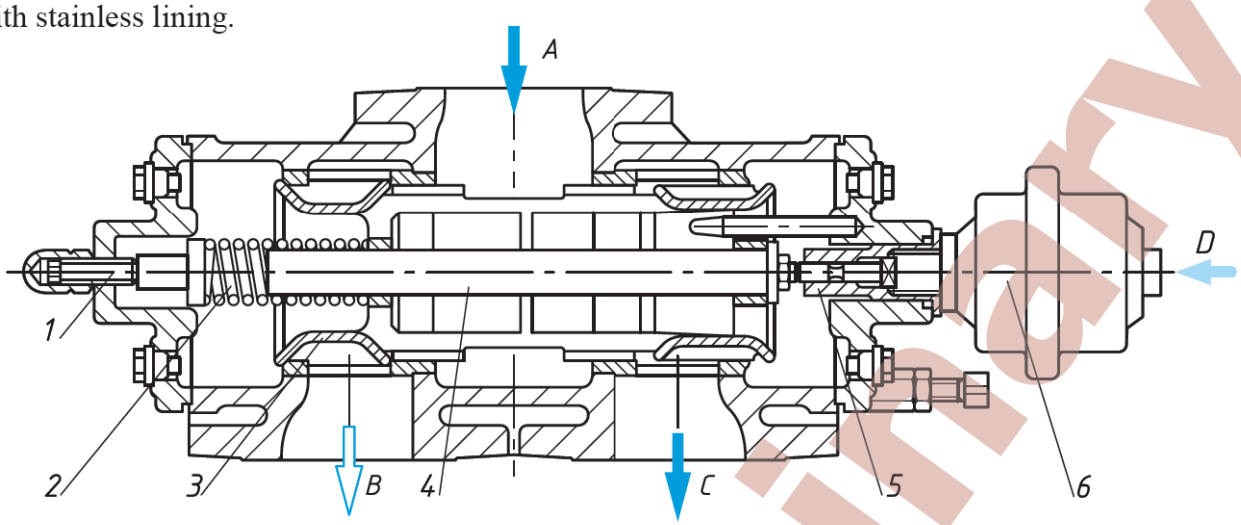


Figure 7.16 – Three-way valve with boost pressure control: *A* – cooling water inlet; *B* – coolant bypassing the cooler; *C* – coolant to the cooler; *D* – air from the inflating receiver; 1 – set screw; 2 – return spring; 3 – spool valve; 4 – spindle; 5 – emphasis; 6 – diaphragm chamber driven by boost pressure (adapted from [15])

In cooling systemsof powerful low-speed engines, as a rule, single-stage centrifugal-axial pumps with double-sided water inlet, with a horizontal or vertical shaft are used (Fig. 7.17 *a*). For engines with moderate power, simpler-designed single-entry pumps are used (Fig. 7.17 *b*).

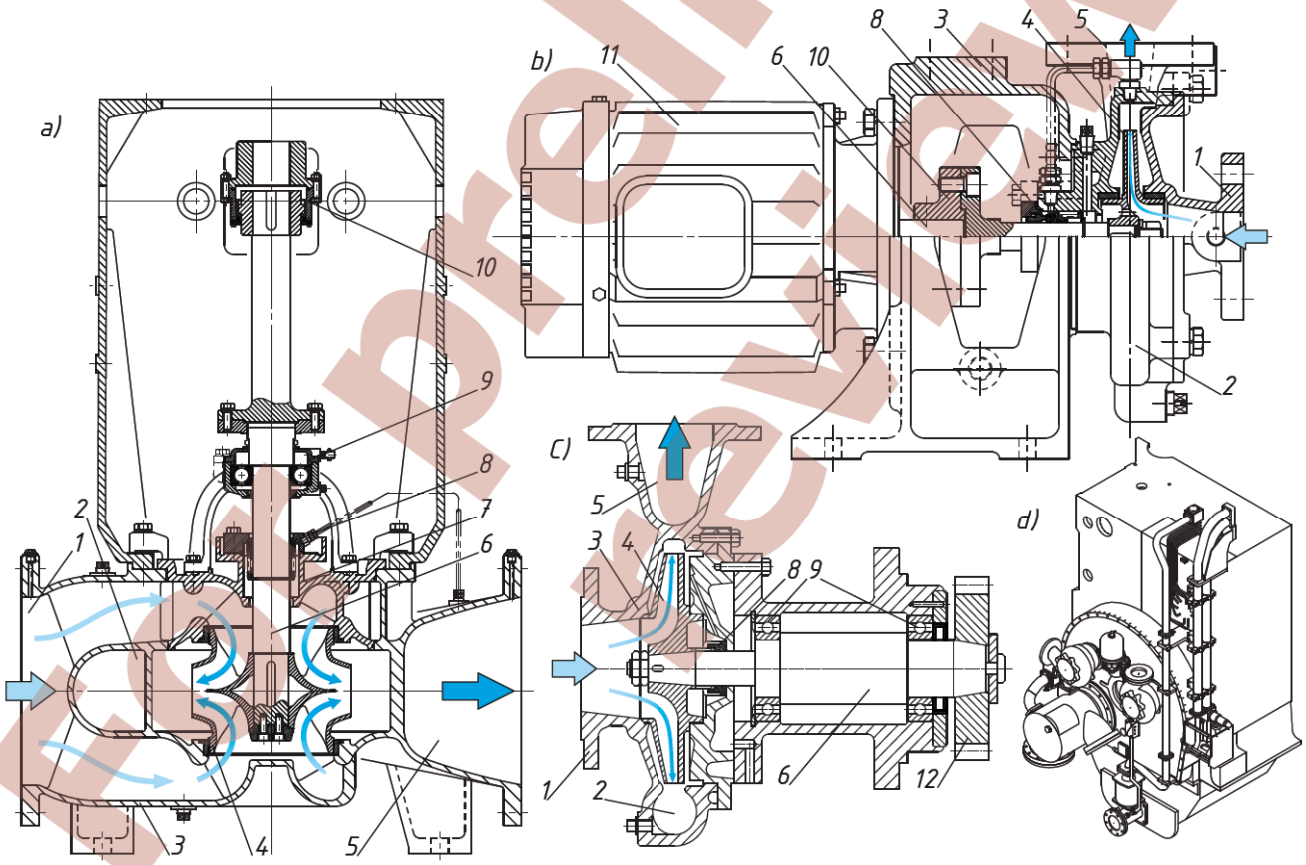


Figure 7.17 – Pumps used in cooling systems of marine engines: *a* – high-capacity centrifugal with double-sided liquid supply and independent drive; *b* – centrifugal with independent drive from an electric motor with one-sided supply; *c* – mounted centrifugal driven by an engine; *d* – location of pumps on the engine drive box: 1 – inlet pipe; 2 – diffuser; 3 – housing; 4 – working; 5 – outlet pipe; 6 – shaft; 7 – cover; 8 – stuffing box seal; 9 – bearings; 10 – drive coupling; 11 – electric motor; 12 – drive gear (adapted from [9, 16, 17, 18])

Preliminary review

Pumps for cooling systemsof low-speed engines have exclusively an electric drive, single-speed or with smooth regulation (using frequency converters). The latter option allows you to match the pump performance to the current engine cooling water needs. The autonomous drive allows for pumping of the cooling system in all engine operating modes, including during stop. All pumps of marine engine cooling systemsare duplicated, which improves the reliability of the system as a whole and simplifies its operation.

On high- and medium-speed engines, along with an autonomous drive, mounted pumps driven by the engine itself are used (Fig. 7.17 *c*). Typically, such pumps are gear driven from a drive box on the nose of the engine (Fig. 7.17 *d*).

A detailed discussion of pump designs is beyond the scope of issues discussed in this book. For a more detailed study of these units, we recommend that you consult specialized literature.

Thermostatic valves.

The principle of regulating the coolant temperature in the cooling circuits is similar to the previously discussed principle of regulating the oil temperature. Thermostatic valves, depending on the temperature of the coolant at their inlet, bypass part of the flow through the cooler, and part of the flow back to the inlet of the circulation pump. By changing the ratio between the flows at the pump inlet, the set temperature is maintained.

Each thermostatic valve of the corresponding circuit is configured to maintain the set temperature in automatic mode.

In cooling systems of marine engines, direct and indirect thermostats are used.

In direct-acting regulators, the thermostatic element directly acts on the housing that regulates the flow of coolant. In indirect-acting regulators, the temperature of the coolant is converted into a signal, most often an electrical one, which is fed to an amplifier, from which the actuator of the temperature regulator receives a control signal that determines the degree of influence of the regulator on the flow.

The general structure and operating procedure of a direct-acting thermostat is shown in Fig. 7.18 *a*, *b*. The fluid flow is controlled using a spool regulator sliding in the housing and driven by two thermostatic elements. Reverse movement is ensured by a return spring. If the temperature in the cooling circuit is insufficient, the spool releases water in a small circle, from the engine again to the pump inlet (Fig. 7.18 *a*). As the temperature rises, thermostatic elements washed by the flow act on the stop and move the spool, redirecting the fluid flow into the cooler (Fig. 7.18 *b*). In the considered design of the thermostat, it is possible to stepwise change the control temperature. To achieve this, the spool drive uses two thermostatic elements with different response temperatures. One of them, a high-temperature one, acts on a rigidly fixed stop, the other, a low-temperature one, acts on a stop connected to a pneumatic cylinder, which, when compressed air is supplied under the piston, can be diverted from the thermostatic element. In this case, the regulator will begin to respond only when the temperature reaches the response threshold of the high-temperature thermostatic element.

The operation of a direct-acting thermostatic element (Fig. 7.18 *c*) is based on the use of the phenomenon of volumetric expansion of paraffin during its phase transition (melting). The transition of paraffin from a solid to a liquid state leads to an increase in its volume by 7...10%. Since hydrocarbons of the paraffin group have different melting points, the use of phase transition makes it possible to create thermostatic elements tuned to different response temperatures.

For a more uniform temperature distribution, copper or aluminium powder is added to the paraffin filler. An airtight brass case is filled with this mixture.When the set temperature is reached, the paraffin melts, which leads to an increase in the volume of the mixture and an effect on the polymer diaphragm, through which the force is transmitted to the spindle. As the paraffin melts (from the moment the melting begins to its completion, the temperature of the mixture increases by 7...10°C), the spindle is pushed out of the housing, driving the actuator (in this case, the spool).

Indirect-acting temperature controllers are used when the force or stroke developed by the thermostatic element is insufficient to move the actuator. The indirect-acting regulator circuit includes a temperature sensor, a device for amplifying the power of the output signal, and an actuator that di-

For preliminary review

rectly acts on the regulator.

Indirect-acting thermostats are divided into pneumatic and electric.

Pneumatic thermostats use compressed air pressure as auxiliary energy. In electric-type thermostats, the control element is moved by an electric motor (Fig. 7.18 *d*). The latter are becoming increasingly widespread on vessels, and the governor control process is increasingly being integrated into the engine control system.

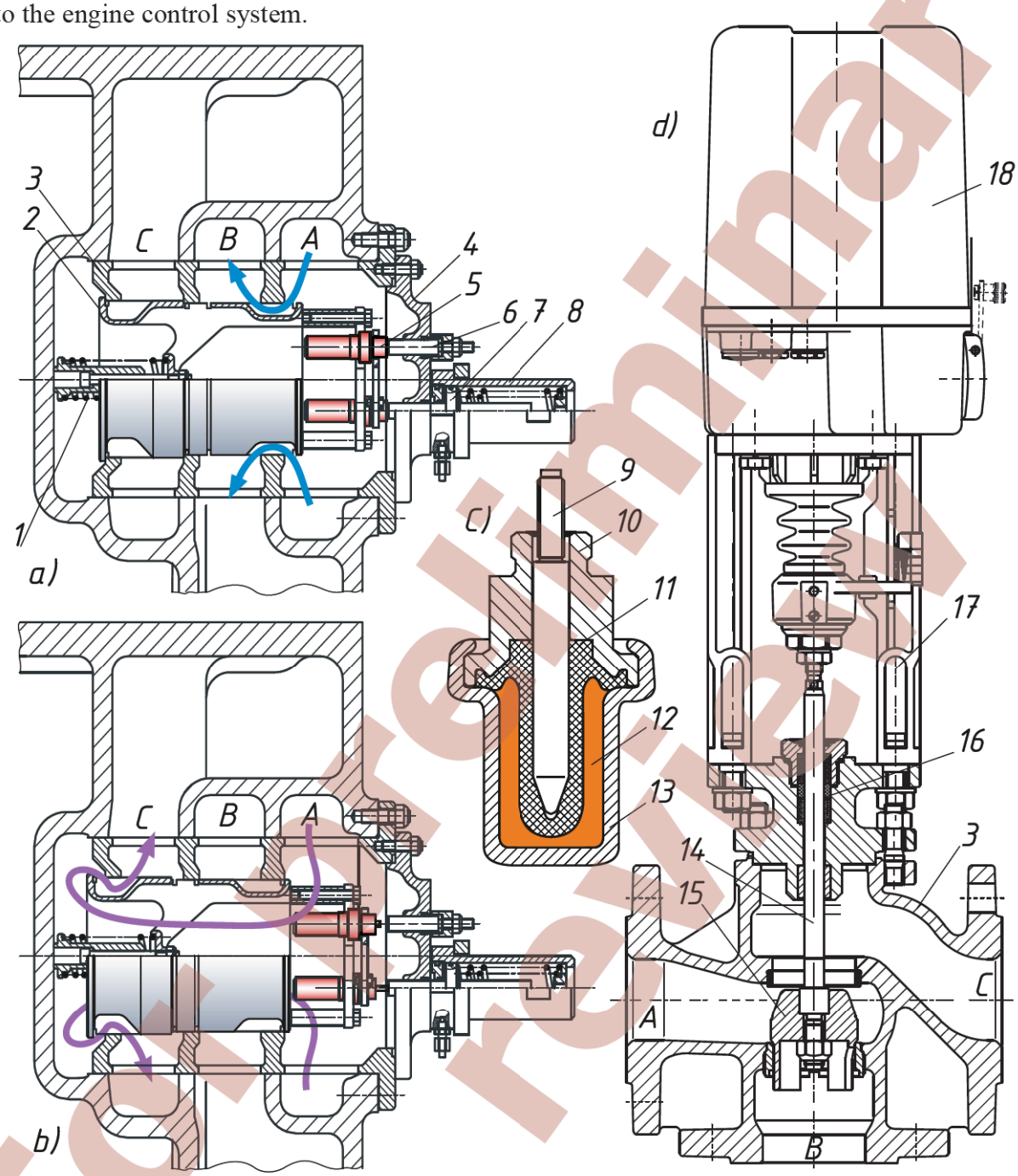


Figure 7.18 – Thermostatic devices; *a*, *b* – design and principle of operation of the thermostatic valve of the 32 series engine from Wärtsilä; *c* – thermostatic element; *d* – three-way control valve with electric drive: *A* – water supply from the engine; *B* – water supply bypassing the cooler; *C* – water supply to the cooler; 1 – return spring; spool-type flow distributor; 3 – valve housing; 4 – cover; 5 – thermostatic element; 6 – adjustable stop; 7 – pneumatic piston with stop; 8 – pneumatic cylinder; 9 – spindle; 10 – top cover; 11 – polymer diaphragm; 12 – stearic filler; 13 – brass case; 14 – spindle; 15 – distribution valve; 16 – seal; 17 – guides; 18 – electric drive (adapted from [19, 20])

Armature. To control the flow of liquids in the cooling system, various fittings are used, which make it possible to disconnect damaged sections of the system by redirecting flows along backup

For preliminary review

lines. Disconnect individual elements from the cooling system so that the rest can function normally. For example, for dismantling cylinder covers, special valves are provided that allow you to disconnect this cover from the system so that you do not have to drain water from the entire system.

The general purpose of the fittings is to turn on or off flows, regulate flow, temperature or pressure of the flow and protect against off-design modes.

According to their purpose, valves are distinguished: shut-off valves (turning on and off flow), control valves (changing or maintaining a given flow rate, pressure, temperature), safety valves (preventing excessive pressure increases, preventing changes in flow direction) and control valves (level indicators). The listed fittings can be installed both on pipelines and on individual units of the cooling system.

According to the control method, a distinction is made between valves with manual, electric, hydraulic, pneumatic drives and self-acting ones, including pulse ones, driven by the medium itself. The most widespread in cooling systems are shut-off valves, valves and taps of various designs (Fig. 7.19), as well as check and safety valves.

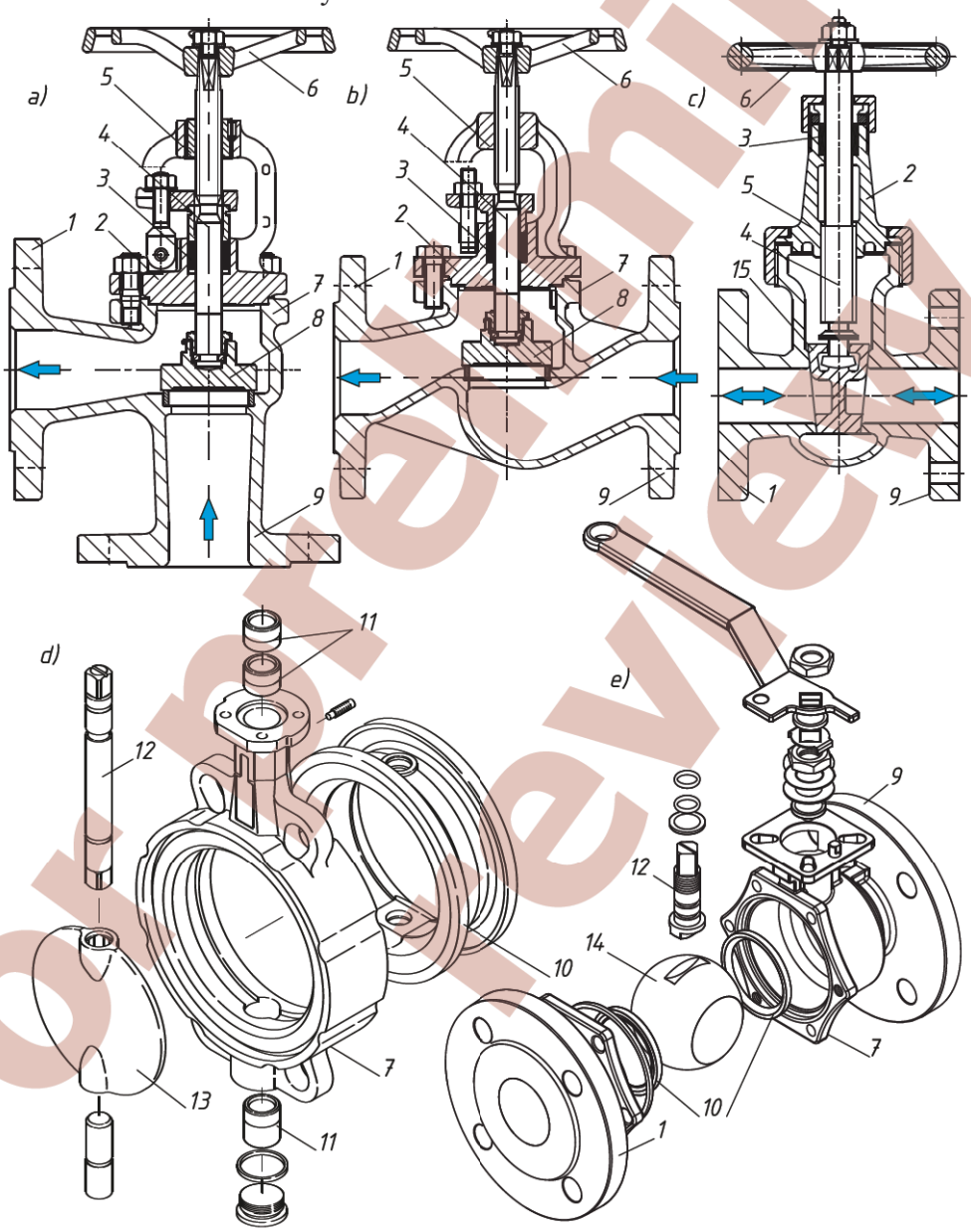


Figure 7.19 – Shut-off devices used in cooling systems of marine engines: *a* – angle non-return shut-off valve; *b* – horizontal non-return shut-off valve; *c* – clinker valve; *d* – butterfly type rotary valve; *e* – ball valve; 1 – output flange; 2 – cover; 3 – stuffing box seal; 4 – lead screw (spindle); 5 – running nut; 6 – flywheel; 7 – housing; 8 – locking plate; 9 – inlet flange; 10 – polymer seals; 11 – shaft bushings; 12 – spindle; 13 – damper; 14 – ball locking element; 15 – wedge valve (adapted from [20, 21, 22])

For preliminary review

In valves, the closing housing sits on the saddle, moving in the direction of flow (Fig. 7.19 *a*, *b*), in gate valves it moves perpendicular to the direction of fluid movement (Fig. 7.19 *c*, *d*) in taps it rotates around its axis (Fig. 7.19 *d*, *e*). In check valves, the closing element is opened by a flow of medium in one direction and closed in the opposite direction. The safety valve opens under the influence of excess (above the established norm) pressure and closes when it is restored.

Valves can be through, in which the direction of movement of the medium at the inlet and outlet does not change (Fig. 7.19 *b*), and angular, in which the direction at the outlet is located at a certain angle (usually 90°) to the direction of the inlet (Fig. 7.19 *a*). The valve shutter is made in the form of a spool (fungus) or a diaphragm, moving with the help of a spindle connected to them.

On pipelines of large diameters, valves are installed as shut-off valves (Fig. 7.19 *c*). The locking part consists of two discs in which a wedge-shaped gate is installed, which is why such gate valves are also called clinkets. The spindle associated with the valve passes through the seal in the valve cover and has an external thread that fits into a sleeve mounted on the bracket. The handwheel moves the spindle, moving the shutter. In the closed position, the wedge-shaped valve expands the disks, pressing them tightly against the sealing rings of the housing passage holes, and blocks the movement of the medium.

Where there is a need to quickly shut off the flow, butterfly type disc shut-off devices are often used (Fig. 7.19 *d*). The ring-type housing contains a rotary disk, which rotates to regulate the flow of liquid or block its movement. These devices are compact, easy to design and maintain, and can be easily integrated into any pipeline, as they can be installed in any position.

In addition to valves and gate valves, ball valves are used as shut-off valves (Fig. 7.19 *e*). The passage through them is blocked by turning a spherical plug with a hole. These devices are one of the types of shut-off pipeline valves, the regulating element in which is capable of rotating around its own axis. Such valves are used in almost all pipelines where reliable shutoff of the flow of the working medium may be necessary. They are distinguished by their short length, light weight, simplicity and reliability.

Coolers. In fresh water cooling circuits, both shell-and-tube type and plate-type coolers are used, the designs of which were discussed by us earlier.

In seawater cooling circuits, priority is given to shell-and-tube type coolers with straight pipes (Fig. 7.20). Such heat exchangers operate with contaminated coolant and require more frequent cleaning than fresh water heat exchangers.

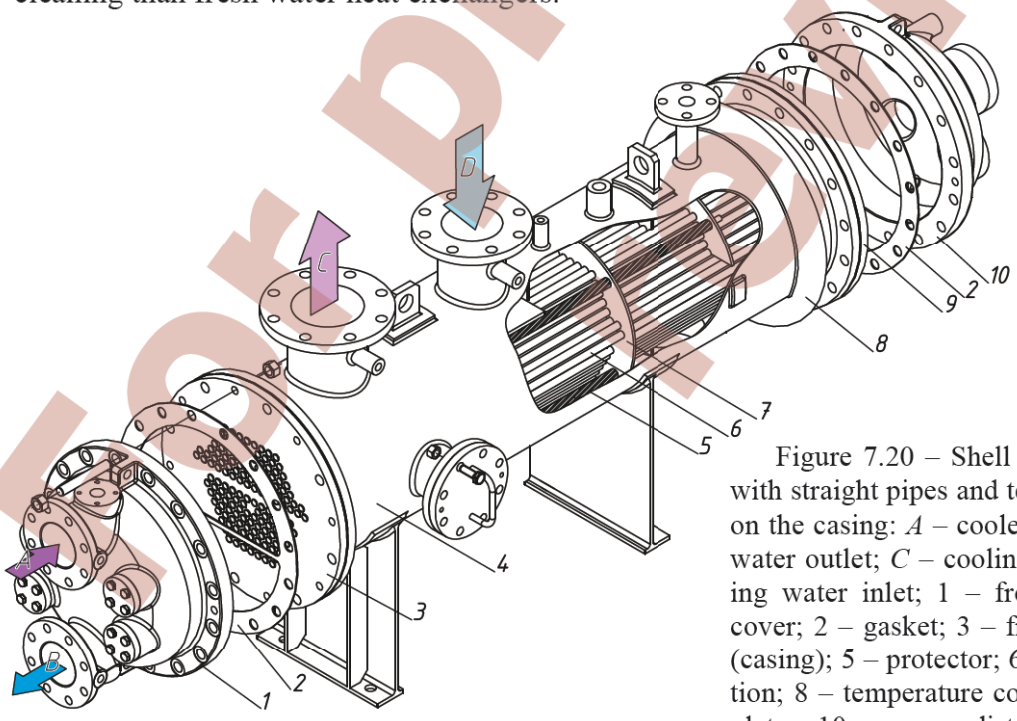


Figure 7.20 – Shell and tube heat exchanger with straight pipes and temperature compensators on the casing: *A* – cooled water inlet; *B* – cooled water outlet; *C* – cooling water outlet; *D* – cooling water inlet; 1 – front distribution chamber cover; 2 – gasket; 3 – front tube plate; 4 – body (casing); 5 – protector; 6 – tube bundle; 7 – partition; 8 – temperature compensator; 9 – rear tube plate; 10 – rear distribution chamber cover (adapted from [23])

For preliminary review

To prevent heat loss to the environment and burns to operating personnel, all heat exchangers, auxiliary equipment and pipelines are covered with a layer of thermal insulation. The insulation density must be such that the temperature on its surface does not exceed 40 °C.

Literature used when writing section 7:

1. Tsalapatis D. Development Aspects. MAN B&W Diesel, Denmark, MAN B&W Diesel Evening, May 15, 2003, Athens. – 72 p.
2. MAN B&W L28/32. General information D10. 1607535-6.6. Holeby Generatinc Sets. – 41 p.
3. Spahni M., Brunner H. Wärtsilä 82 Low-Speed Engine Types. High-Economy Engines for Panamax Containerships, Large Tankers and Large Ore Carriers. Wärtsilä Switzerland Ltd, Winterthur. 2008. – 15 p.
4. Demmerle R. Sulzer RTA84C and RTA96C engines: The reliable driving forces for large, fast containerships. Wärtsilä NSD Switzerland Ltd, Winterthur. 1997. – 51 p.
5. Geist M. Sulzer RTA-8T engines: Compact two-strokes for tankers and bulk carriers. Wärtsilä NSD Switzerland Ltd, Winterthur. 1998. – 53 p.
6. Stanglmaier R. An innovative glass-metal coating to provide corrosion resistance and a thermal barrier for highly loaded engine components. / Stanglmaier R., Gross T., Märkisches G., Veriotski V., R. Conradt // CIMAC Congress 2007, Vienna Paper No. 184. – 10 p.
7. Wärtsilä 18V34SG. Instruction Manual. Wärtsilä Finland Oy Vaasa Factory. Finland. – 416 p.
8. Bilousov, I., Bulgakov, M. & Savchuk, V. Modern Marine Internal Combustion Engines. Springer International Publishing. 2020. – 385 p. https://doi.org/10.1007/978-3-030-49749-1
9. Engine type W46B Instruction Manual. Wartsila Finland Oy Vaasa Factory. Finland. 2011. – 458 p.
10. WinGD W-X82. Operation Manual «Marine». 2017 Winterthur Gas & Diesel Ltd. Winterthur, Switzerland. 2017. – 500 p.
11. Instructions for Main Engine. Type S46MC-C. Volume III-Component sdescriptions, Edition 2. MAN B&W Diesel A/S. Denmark. – 168 p.
12. Technical Documentation. Engine V 32/40, Spare Parts Catalogue. 6633-3, MAN B&W Diesel AG, Augsburg, Germany. 2003. – 422 p.
13. Operation & Maintenance Manual. Marine diesel engine L136, L136T, L136T1, L086T1. Marine generator engine. AD136, AD136T. AD13ET1, ADQ86TI. 65.99897-8080A, DOOSAN Infracore Co., Ltd. 2003. – 180 p.
14. Technology Review. Wärtsilä NSD Finland Oy WNS00M05E, Vaasa, Finland. – 28 p.
15. Operating Instructions TBD 645. 0299 6391 en, Motoren-Werke Mannheim AG, Mannheim. Germany. – 244 p.
16. Engine operating instructions Type M32. AA000439, Caterpillar Motoren GmbH & Co. KG, Kiel, Germany. 2003. – 1229 p.
17. Centrifugal pump, Finished DWG. Taikokikai industries CO., LTD. Yamaguchi, Japan. 2008. – 16 p.
18. Customer 3000008396. Sungdong Shipbuilding & Marine Engineer ING CO., LTD. Yamaguch, Japan. 2008. – 160 p.
19. Control valve – 3-way with flanges – in mixing construction. ARI-Armaturen Albert Richter GmbH & Co.KG, 33756 Schlo Holte-Stukenbrock. Edition 01/06. – 96 p.
20. Shut-off valves. Type: V3NBC-316-…,V1-316-299-015. D-19061. Operating and installation instructions. Stop valve with gland seal Stobu Kuwo GmbH, Schwerin, Wüstmark. – 12 p.
21. Wafer pattern butterfly typ Z 011-A. Ebro-Armaturen. 2006. – 6 p.
22. Ball Valves. Operation and Installation Manual. END-Armaturen GmbH & Co. KG Oberbecksener Straße 78. D-32547 Bad Oeynhausen. 2002. – 28 p.
23. Instruction manual. Cooler. DongHwa Entec. Busan, Korea. – 11 p.

For preliminary review

SECTION 8

Starting and reversing systems

8.1 Engine starting process

A feature of the operating process of piston engines is that the working stroke in the cylinder must necessarily be preceded by the compression stroke, and in diesel engines the charge temperature at the end of compression must be sufficient for self-ignition of the supplied fuel. To fulfill these two conditions, when starting the engine, it must first be spun up using an external energy source. This is the function that the engine starting system performs. If the engine is reversible, the starting system must ensure that the engine starts in both directions, in which case it is called a start and reverse system.

8.2 Electric starting

On small engines that are used as emergency diesel generators or to drive lifeboats, starting with electric starters is widely used (Fig. 8.1).

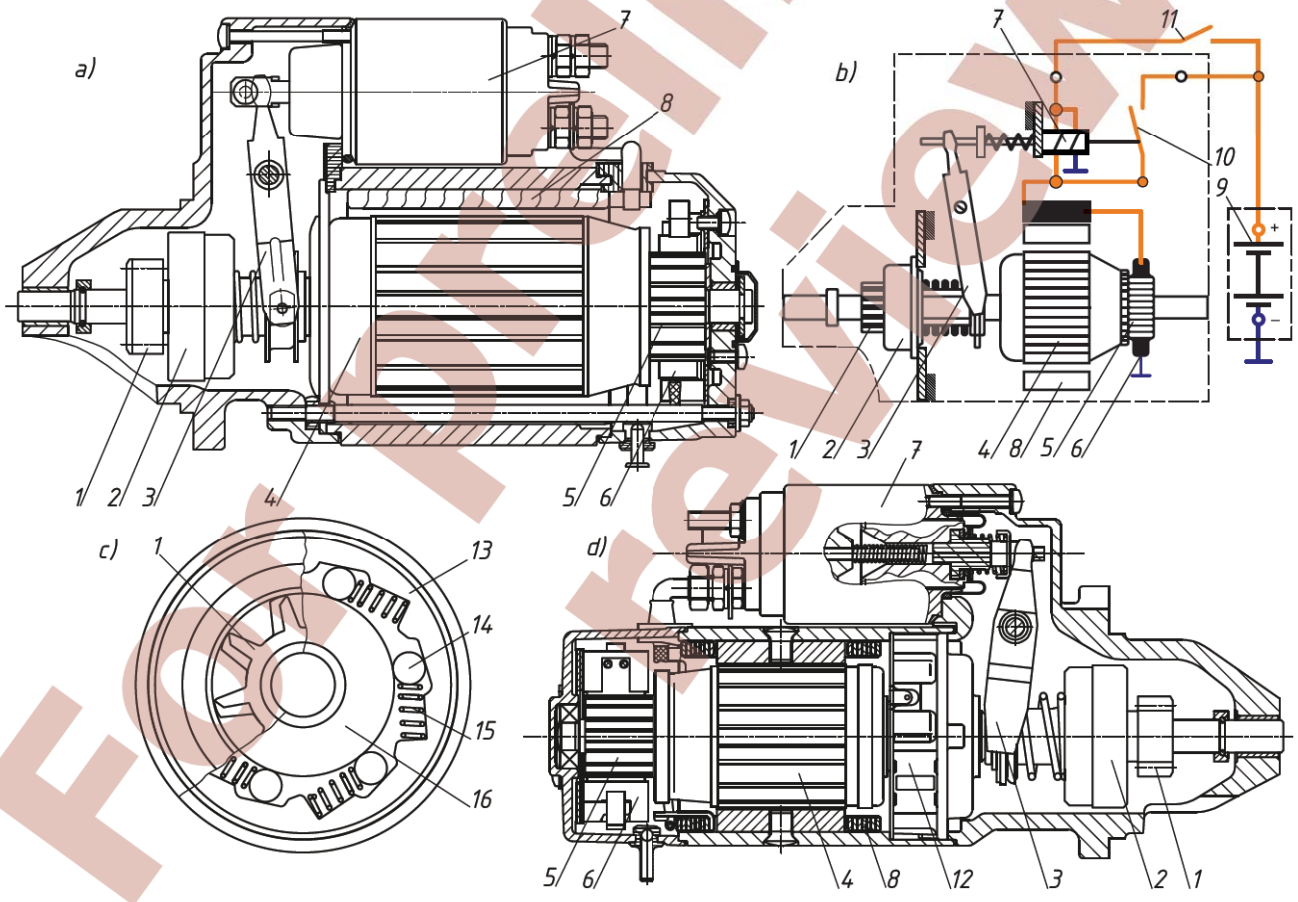


Figure 8.1 – Electric starter with direct drive (*a*) and its connection diagram (*b*), roller safety clutch (*c*), electric starter with reduction gear (*d*): 1 – engagement gear with ring gear on the flywheel; 2 – roller safety clutch; 3 – drive lever for engaging the gear; 4 – rotor; 5 – collector; 6 – brush mechanism; 7 – solenoid relay; 8 – stator winding; 9 – battery; 10 – central switch of the retractor relay; 11 – starter switch; 12 – planetary gearbox; 13 – safety clutch stator; 14 – rollers; 15 – spring; 16 – safety clutch rotor (adapted from [4])

For preliminary review

The electric starter is a DC commutator engine, with direct drive (Fig. 8.1 *a*) or driven through a reduction gear (Fig. 8.1 *d*), powered by a starter battery (Fig. 8.1 *b*). Starter engines have a special design with four brushes, which increases the armature current density and starter power.

When the starter is turned on, electric current (through the start relay) is supplied to the solenoid coil of the solenoid relay. The solenoid core retracts and, through a linkage, engages the starter engine gear with the ring gear on the engine flywheel. After this, the contacts of the starter relay close, connecting it to the battery and switching power to the holding winding of the solenoid. The relay contact group is designed for high starting current, reaching hundreds of amperes. To protect the electric motor from excessive spinning after starting the engine, the starter gear is connected to the shaft through a ratchet or roller-type overrunning clutch (Fig. 8.1 *c*), which ensures one-way transmission of mechanical energy from the starter to the flywheel. If the flywheel speed increases above the starting speed, the clutch wedges and ensures free rotation of the gear without transmitting torque to the electric motor shaft. After the starter is turned off, the drive gear returns to its original position.

Electric starting is relatively simple and is especially convenient when remote controlled, but starter batteries are heavy, difficult to maintain, have a relatively short service life and high cost.

8.3 Pneumatic systems for starting and reversing engines

On marine engines, compressed air energy is used as the main type of starting energy.

There are two main ways to air start engines.

Direct launch. Compressed air is supplied directly to the working cylinders of the engine in accordance with their operating order. Air pressure forces the pistons to move from TDC to BDC, which causes the crankshaft to turn. This starting method has no power limitations and is therefore widely used on both low-speed and large medium-speed engines.

Starting with a pneumatic starter. Compressed air is used to drive a pneumatic engine, which, at the time of start-up, is engaged with the ring gear on the flywheel, driving the diesel crankshaft. This starting method is used primarily for engines of medium and low power.

8.3.1 Pneumatic starters

Pneumatic starters are distinguished by a wide variety of designs, both in layout schemes and in methods of converting compressed air energy into mechanical work. They have a number of advantages over other starting methods, which in many cases make their use preferable for economic and technical reasons.

These benefits include:

- simplicity of the starting system, which does not require intervention in the design of the engine itself;
- ease of regulation of rotation speed and torque;
- the possibility of full braking under load without compromising the design and performance characteristics of the pneumatic engine;
- no overheating;
- large resource of work;
- complete fire and explosion safety;
- insensitivity to adverse environmental factors (dust, moisture, etc.).

Conventionally, all existing designs of pneumatic starters can be divided into two main types: with positive-displacement or dynamic-action engines.

8.3.1.1 Pneumatic starters with positive displacement engines

Gear pneumatic engines are widely used as positive displacement engines in starters, in which compressed air acts on two gear rotors rotating in a stator tightly fitted to them.

Gear pneumatic engine. In the air starter shown in Fig. 8.2, one of the gear rotors transmits rotation to an internal gearbox. This allows you to increase the torque on the shaft. The second rotor rotates freely on supports installed in the housing. Compressed air supplied to the working chamber acts on the side surfaces of the gear teeth. In this case, forces arise equal to the product of com-

For preliminary review

pressed air pressure and the area of the lateral surface of the tooth, which cause rotation of the gears, one of which rotates clockwise, and the other in the opposite direction.

Gear pneumatic engines are simple in design, can easily withstand overloads, and are reliable due to the small number of parts. Their durability is determined by the same conditions as the durability of conventional gears. Disadvantages include high noise during operation, large weight and dimensions, and increased specific consumption of the working fluid. The latter is due to the fact that gear engines of this type practically do not use the expansion energy of compressed air.

Air is supplied to the starter air engine through a starting valve, which is actuated by a starting piston connected to the starting control system. Simultaneously with the supply of air to open the starting valve, control air enters the pneumatic cylinder of the retractable shaft drive. Moving along the axis, the retractable shaft engages a gear (not shown in the figure) with the ring gear on the engine flywheel.

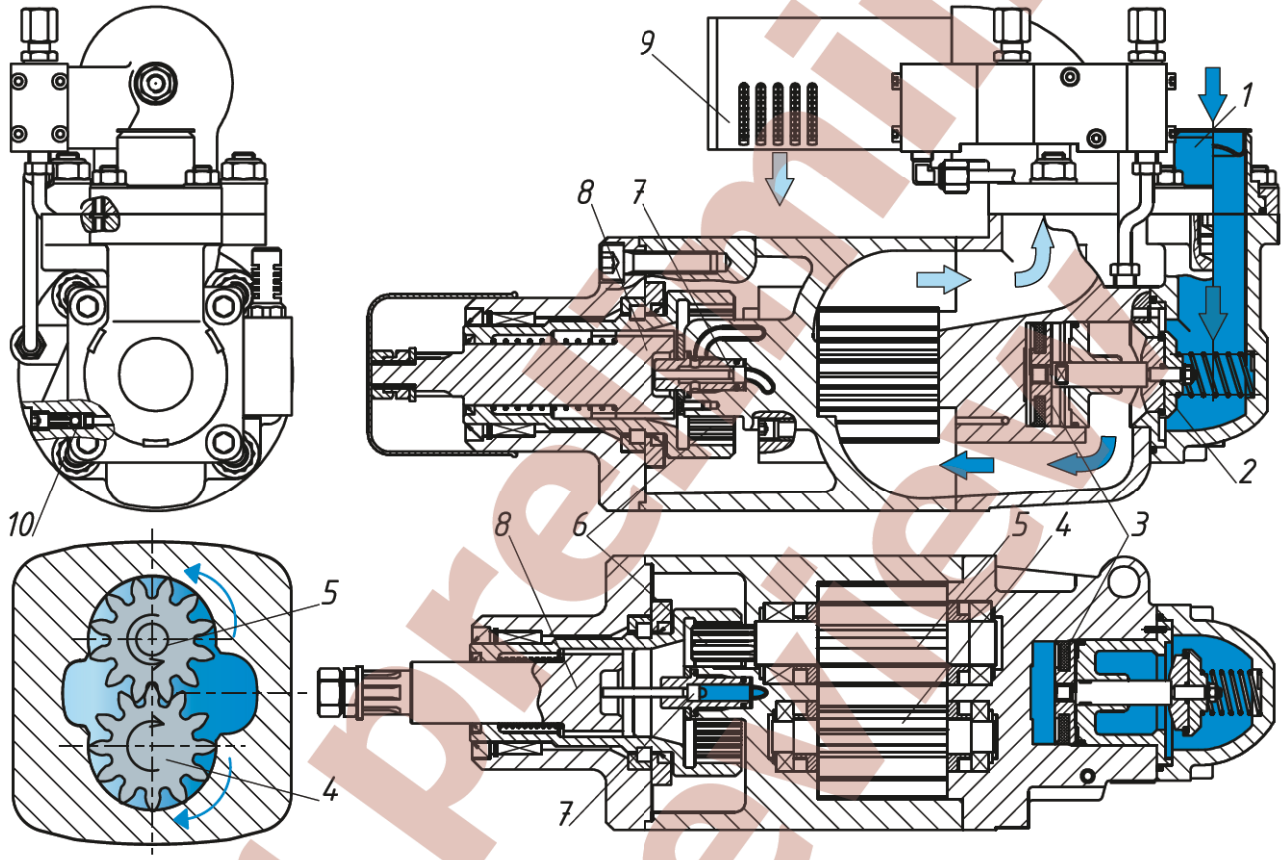


Figure 8.2 – Air starter with a gear pneumatic engines: 1 – compressed air inlet; 2 – start valve; 3 – pneumatic drive of the starting valve; 4 – secondary rotor; 5 – primary rotor; 6 – gearbox with internal gearing; 7 – pneumatic cylinder for the drive of the retractable shaft; 8 – retractable shaft; 9 – exhaust air outlet to atmosphere; 10 – safety valve (adapted from [5])

Vane air engines. Another type of positive displacement air engine used in air starters is plate (vane) expansion machines (Fig. 8.3).

In machines of this type, due to the eccentric location of the rotor in the cylindrical stator, a crescent-shaped cavity is formed between them. The radial grooves of the rotor contain movable plates, which, under the action of the centrifugal force arising when the rotor rotates, move out of the grooves and are tightly pressed against the inner cylindrical surface. Rotating plates divide the space between the rotor and stator into working chambers, the volume of which changes as the rotor turns. When compressed air is supplied to the working chamber of the pneumatic engine, forces arise that act on the plates that limit the volume of the chamber. Due to the eccentric location of the rotor relative to the stator, the areas of the plates are different, and therefore the forces acting on

For preliminary review

them also differ in magnitude. After passing the point of maximum release of the plates from the grooves, the volumes of the working chambers begin to decrease. In this area there is a hole for exhaust air discharge. From the resultant of all applied forces, a torque arises, leading to rotation of the rotor, during which the volumes of some of the working chambers increase, due to which the compressed air contained in these chambers expands. The expansion work performed in this case is converted into additional mechanical energy of rotor rotation.

A single-stage gearbox allows you to increase the torque on the starter shaft, and the friction clutch protects it from overloads.

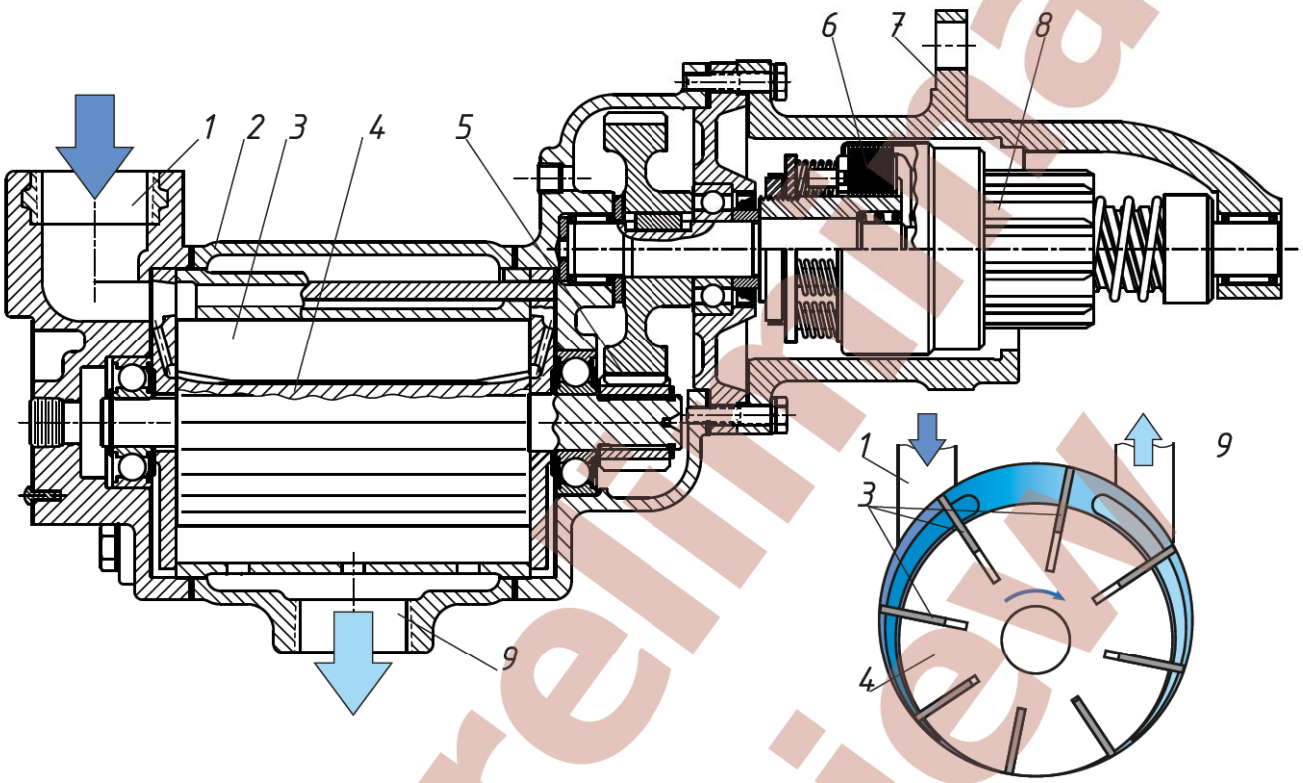


Figure 8.3 – Air starter with a plate (vane) pneumatic engine 150BMG: 1 – compressed air inlet; 2 – housing; 3 – gate; 4 – rotor; 5 – gearbox; 6 – safety clutch; 7 – mounting flange; 8 – retractable shaft with drive gear; 9 – exhaust air outlet to atmosphere (adapted from [6])

8.3.1.2 Pneumatic starters with dynamic engines

On high-power engines, pneumatic starters are used, equipped with axial or radial air turbines as an engine.

In Fig. 8.4 shows a cross-section of a starter equipped with a double-sided radial turbine, in which the impeller with the output shaft rotates on two ball bearings built into the housing. The impeller blades are located peripherally, as well as the fixed stator guide blades, mounted on two semi-casings. Compressed air enters the peripheral gap between the stator semi-housings and is supplied to the inter-blade channels of the impeller, through which the direction of its movement changes. As a result, a force is applied to the blades and creates a torque. In this case, the compressed air gives up part of its energy to the turbine wheel and exits into the internal cavity of the stator, from where it is removed into the atmosphere through the exhaust valve. Turbine pneumatic engines using radial or axial turbines make it possible to obtain very high rotational speeds (20,000...40,000 min^{-1}) of the output shaft and low torques. This requires the installation of a more complex planetary gearbox and devices for protecting it from overloads (friction clutches).

The main disadvantages of turbine pneumatic engines made on the basis of classical radial and axial turbines are the complexity of manufacturing due to the presence of specially profiled nozzles and working blades of complex shape, high speed, and reduced efficiency at low flow rates of the working air.

for preliminary review

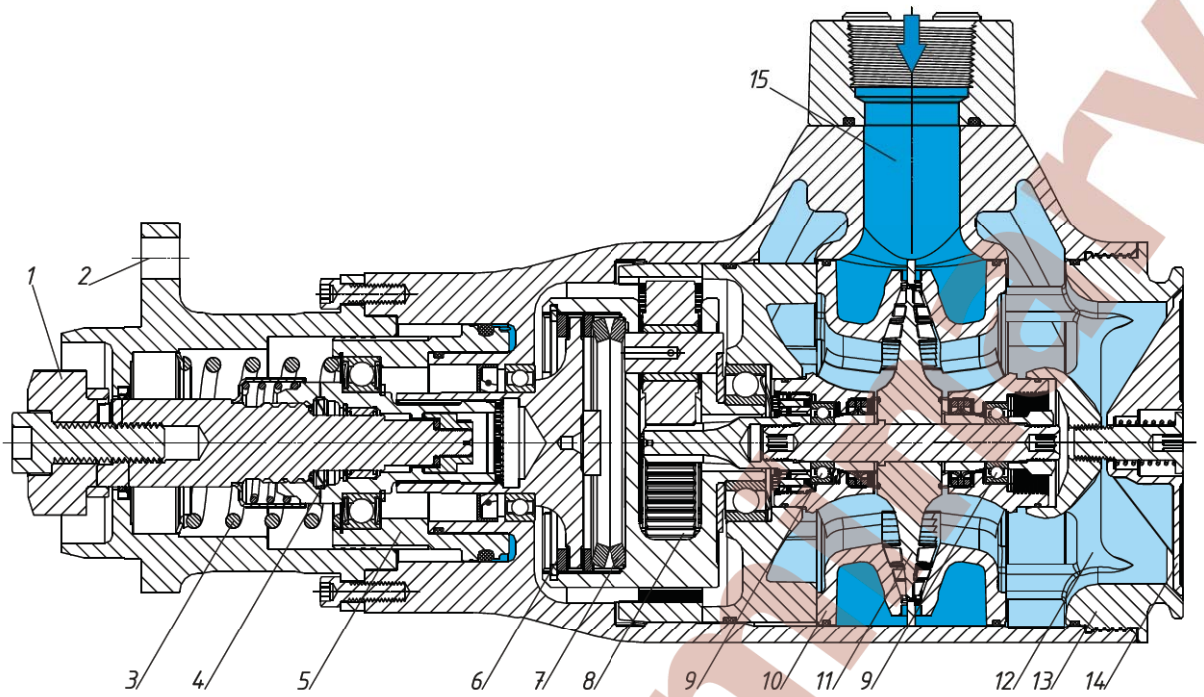


Figure 8.4 – Air starter with a turbine drive of the ST600 series: 1 – gear for engagement with the gear wheel on the flywheel; 2 – connecting flange; 3 – return spring; 4 – safety ratchet mechanism; 5 – drive piston of the mechanism for introducing the drive gear into engagement with the flywheel; 6 – safety friction clutch; 7 – disc spring of the friction clutch; 8 – planetary gearbox; 9 – turbine rotor bearings; 10 – radial turbine stator; 11 – turbine rotor; 12 – air release cavity; 13 – back cover; 14 – exhaust air release valve; 15 – compressed air supply pipe (adapted from [7])

Some manufacturers produce unified starters, which can be equipped with different types of air engines depending on needs.

Regardless of the types of starters, the diagrams for connecting them to the compressed air system differ little from each other (Fig. 8.5).

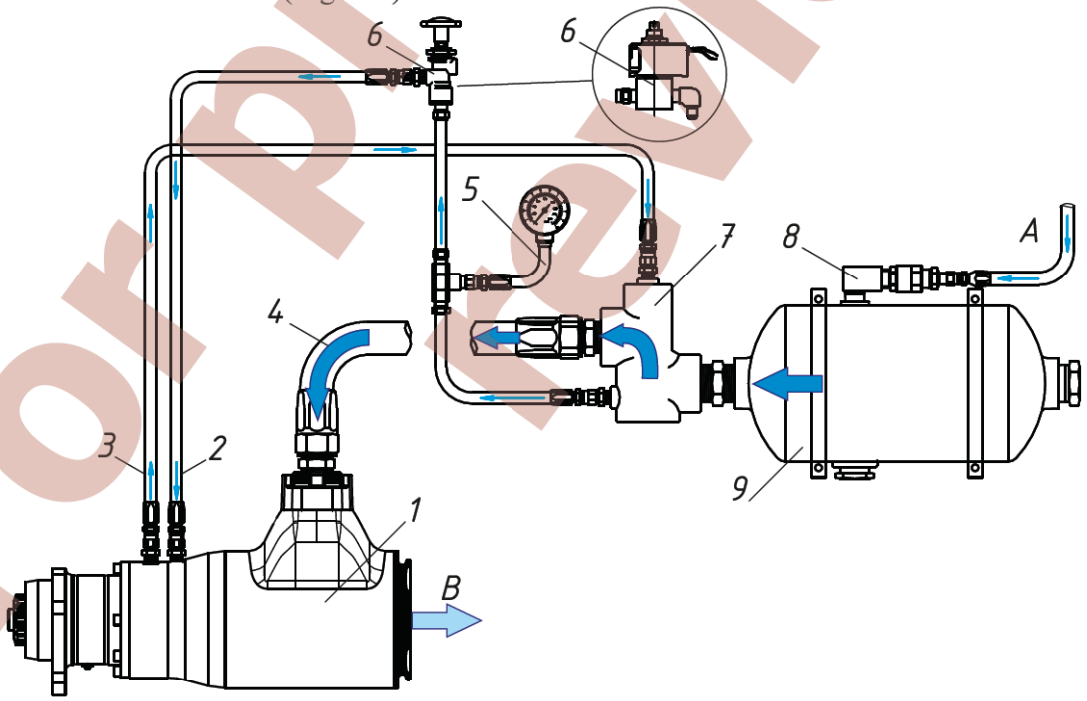


Figure 8.5 – Air starter connection diagram: 1 – air starter; 2 – control air supply line; 3 – control air exhaust line; 4 – air supply to the power drive; 5 – control pressure gauge; 6 – starting valve (local or with an electric drive for remote starting); 7 – central distributor; 8 – air receiver input fittings; 9 – air receiver; A – air from the compressor; B – exhaust air outlet to atmosphere (adapted from [8])

For preliminary review

In any circuit, two circuits can be distinguished, the power air circuit and the control circuit. If the power of one starter is insufficient to reliably start the engine, several of them can be installed – from two to three, connected in parallel. This allows not only to increase starting power, but also to increase the reliability of the starting system by redundant units of the same type.

8.4 Direct starting systems

The direct air starting system of a marine internal combustion engine is used to start, stop (in an emergency) or reverse the engine.

The starting system elements are mounted directly on the engine and provide compressed air to the working cylinders in accordance with the in accordance with the order of their work. Air enters each cylinder under a pressure of 3...1.5 MPa at an angular interval of 2...110° after TDC, which corresponds to the expansion stroke. For four-stroke diesel engines, starting from any position of the crankshaft is ensured if there are at least six cylinders; for two-stroke diesel engines, at least four. Starting with compressed air can be done both with simultaneous supply of fuel to the cylinders (mixed start) and without it (separate start). The supply of compressed air on board must provide at least six consecutive starts for non-reversible engines and 12 starts for reversible engines. Particularly difficult starting conditions are created in main marine diesel engines with direct transmission to a fixed-pitch propeller, since the energy of the starting air must overcome not only the forces of resistance to the rotation of the engine itself, but also the resistance to rotation of the propeller with the masses of water attached to it.

Modern engines predominantly use separate starting, which is preceded by slow cranking of the engine, for which a separate circuit is provided in the starting system. Slow cranking is necessary to monitor the condition of the mechanical part of the engine, check that coolant does not enter the cylinders due to their depressurization, etc. Engine automation features include a slow crank mode when starting the engine after stopping for more than 30 minutes. If after several revolutions of the crankshaft no problemsare detected, the start mode is automatically activated.

The startup process can be divided into three stages:

- cranking of the engine in the initial period under the influence of starting air pressure entering the cylinder, the piston of which was in the starting position;
- subsequent acceleration of the engine under air pressure entering the remaining cylinders in accordance with the order of their operation;
- transition of the engine to operation on fuel.

To shut off the fuel supply during engine acceleration on positive displacement fuel pumps of low-speed engines, a bypass valve with a pneumatic drive is provided. On medium-speed engines, at the moment of starting, the fuel pump racks are switched to zero flow. In electronically controlled engines, the supply is switched off by the central control unit.

A schematic diagram of the direct starting system for medium-speed engines 46 series from Wärtsilä is shown in Fig. 8.6.

The air line from the starting air receiver is equipped with a shut-off valve and a check valve, as well as a purge valve installed in front of the main starting air valve.

The main start valve and the slow rotation valve are actuated by control air supplied through solenoid control valves mounted on the local instrument panel by pressing the start button or by activating the solenoids through the remote control system. Slow cranking of two turns is activated automatically if the engine has been stopped for more than 30 minutes. During slow rotation, air enters the slow rotation valve through the regulator, which reduces the pressure to the required level.

A check valve in the slow rotation line prevents air leakage during main startup.

The shut-off valve prevents air leakage when cranking slowly.

After activating the main start valve, the air is divided into two streams, power and control. The first is fed through flame arresters to the starting valve on the cylinder cover. The second air flow is supplied to the starting air distributor, which sequentially supplies control air to the starting valves

For preliminary review

of the working cylinders in accordance with their operating order. The starting air distributor controls the timing and duration of opening of the starting valves.

The locking valves on the turning gear prevent the engine from starting when the turning gear is engaged.

The main starting valve is used to carry out multiple starts with open isolation valves on the starting air cylinders and to unload the starting line after the start is completed. Marine diesel engines use different designs of main starting valves. In Fig. 8.7 *a*, shows a general view of the main starting valve of engines of the MC and ME series from MAN.

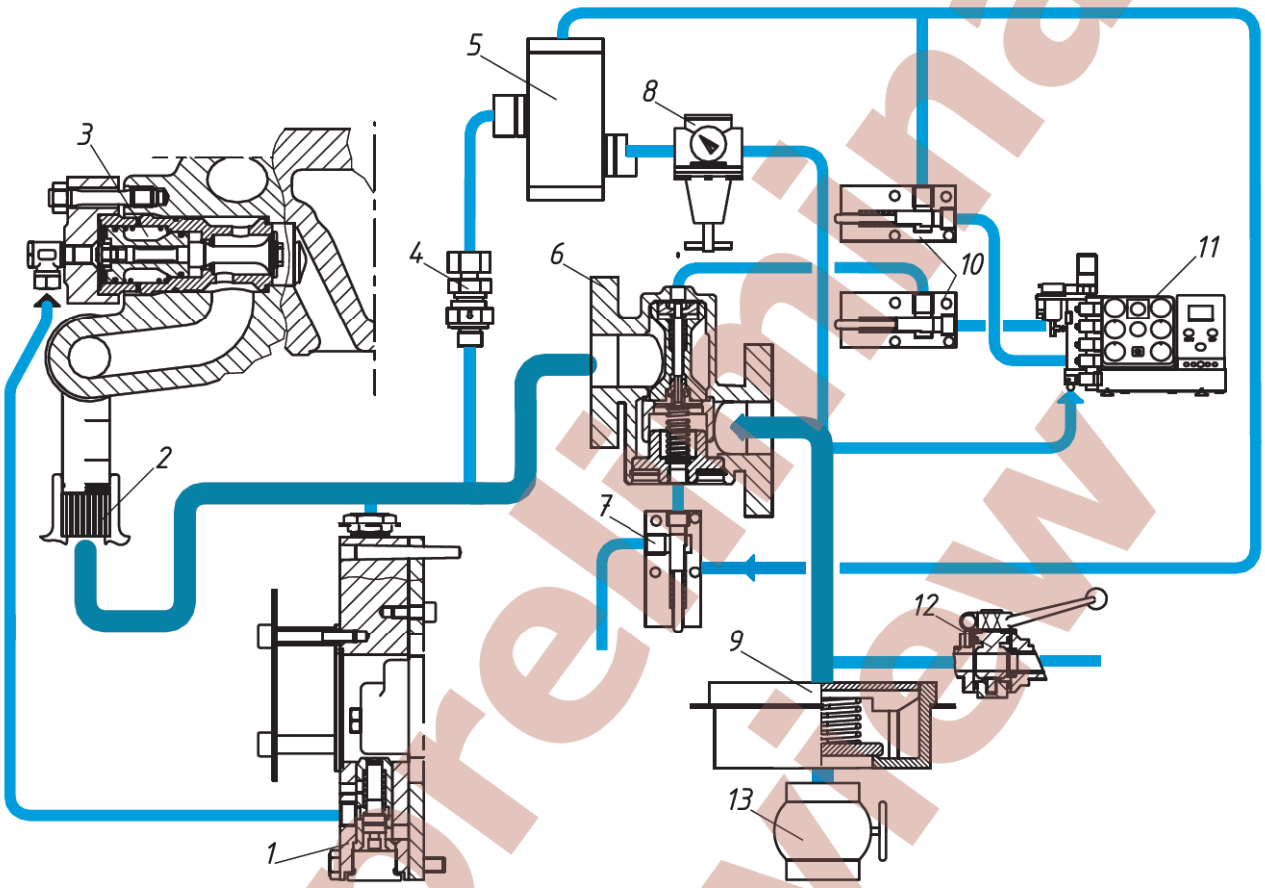


Figure 8.6 – Diagram of the air starting system for medium-speed engines of the L/V 46 series from Wärtsilä: 1 – air distributor; 2 – flame arrester; 3 – start valve; 4 – check valve of the slow cranking system; 5 – slow rotation valve; 6 – main starting valve; 7 – shut-off valve; 8 – pressure regulator; 9 – check valve; 10 – blocking valves; 11 – local control station; 12 – purge valve; 13 – valve on the compressed air receiver; 14 – check valve (adapted from [9])

It is made in the form of a unit consisting of three main elements; starting air valve, slow cranking air valve and check valve. The air supply valves for starting and cranking the engine are of the same type and differ only in size. Structurally, these are ball valves (Fig. 8.7 *b*) with an electric drive of the shut-off device. The valve for cranking the engine is installed as a bypass valve on the bypass channel of the main valve. In addition, the design provides a locking device and a rotation protection system when the turning mechanism is turned on.

In Fig. 8.7 *c*, the design of the main starting valve for medium-speed engines of the L/V 32 series from Wärtsilä is shown. The valve consists of a locking device in the form of a disc and a pneumatic drive mechanism equipped with a control valve. The control valve allows the main valve to be opened from a local control station or remotely. Control air is taken from the main line through a special channel made in the valve housing. The control valve can be activated manually using a button mounted on it, or by compressed air using a solenoid valve designed for remote or automatic

For preliminary review

start. When the valve is operated, air enters the cavity of the power piston, which transmits its effort through the valve stem directly to the main valve and opens it, overcoming the force of the return spring and the pressure in the system. Pressure under the main shut-off device helps keep it in the closed position, ensuring a tight seal.

The starting air distributor controls the supply of control air to each starting valve mounted on the cylinder covers. As a rule, valves of the spool type, disk or cylindrical are used.

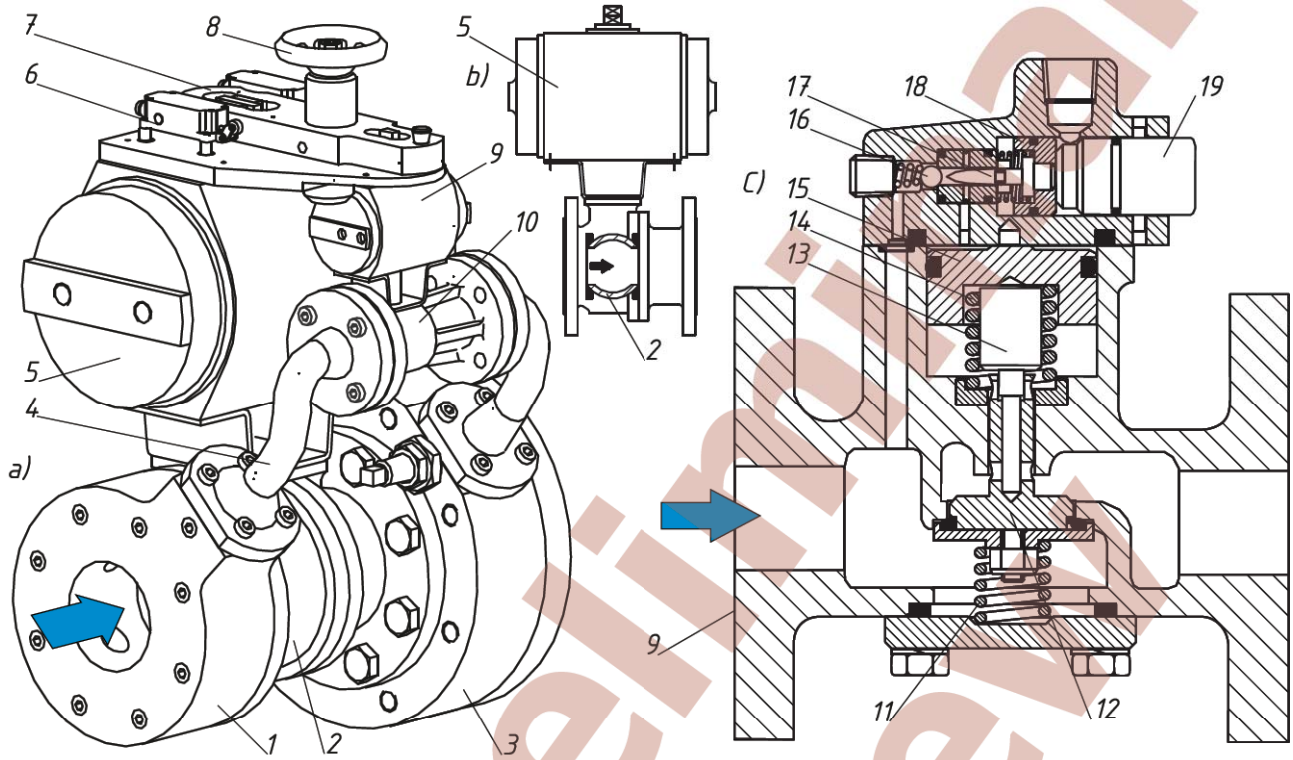


Figure 8.7 – Main starting valves of low-speed engines of the MC and ME series from MAN (*a*, *b*) and medium-speed engines of the L/V 32 series from Wärtsilä (*c*): 1 – inlet device; 2 – ball valve of the main air supply; 3 – check valve; 4 – bypass channel; 5 – electric drive of the main ball valve; 6 – limit switches of the control and protection system; 7 – fixing bar; 8 – locking handwheel; 9 – electric drive of the bypass ball valve; 10 – bypass ball valve; 11, 14 – return spring; 12 – valve plate; 13 – valve stem: 15 – power piston; 16 – control valve; 17 – control valve spindle; 18 – control valve drive piston; 19 – local engine start button (adapted from [9, 10])

In Fig. 8.8 shows the general view (Fig. 8.8 *a*) and the design of the disk distributor used on low-speed engines. It consists of a distribution disc (disc rotor) and an intermediate disc (disc stator), which are located in the distributor housing (Fig. 8.8 *b*, *c*). The camshaft disc is driven into rotation by the engine camshaft through a gear transmission (Fig. 8.8 *b*). Its drive is synchronized with the engine valve timing.

Short and long sectoral recesses are milled on the contact surface of the distribution disk (Fig. 8.8 *d*). Through a short recess, control air is supplied from the distributor housing to each starting air valve, on the cylinders in accordance with the order of their operation. The long recess vents the starting valves, to which no control air is currently supplied. A long recess through holes in the intermediate disk and housing communicates with the atmosphere. Each outlet in the distributor hous-ing is connected by pipes to the corresponding starting air valve (Fig. 8.8 *c*). The control air pressure opens the starting valve and air enters the engine's slave cylinder, causing the piston to move and crank the crankshaft. As the crankshaft begins to rotate, the distributor plate also rotates, thereby moving a short notch from one hole to the next hole. At the same time, control air is supplied to the start valve connected to this hole, and the start valve opens. At the same time, a long recess in the control disk is positioned opposite the opening of the actuating valve, which was activated immediately before, and the control air from this actuating valve is released into the atmos-

For preliminary review

phere, causing the valve to close. When the required crankshaft rotation speed is reached, the fuel supply to the injectors is turned on and the air supply to the distributor is turned off. The engine continues to run on fuel.

The distributor shaft rotates in a bushing mounted in a hole in the center of the housing. Oil from the engine lubrication system is supplied to the bushing under pressure. After starting the engine and stopping the air supply to the distributor, the distribution disc is moved away from the intermediate disc by a small amount. This occurs due to the difference in forces caused by the difference in shaft diameters at opposite ends of the sleeve (Fig. 8.8 *c*). When the engine starts, air pressure presses the distribution disc against the intermediate disc.

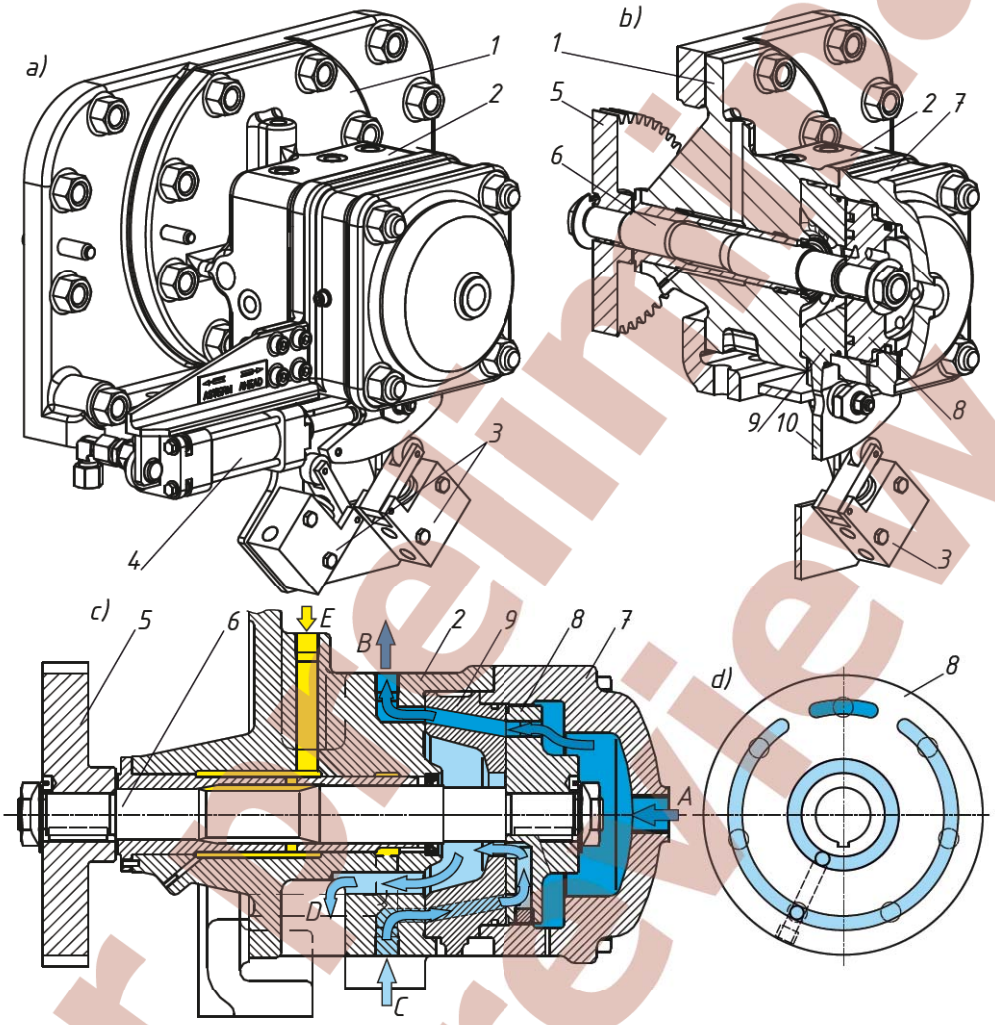


Figure 8.8 – Disc compressed air distributor among cylinders; general view (*a*), general arrangement (*b*); flow diagram (*c*) and type of distribution disk (*d*). 1 – mounting flange; 2 – housing; 3 – limit switches of the alarm system; 4 – pneumatic cylinder of the reverse system; 5 – drive gear; 6 – shaft; 7 – cover; 8 – distribution disk (rotor); 9 – distributor stator; *A* – air of the main starting valve; *B* – air to the starting valves of the working cylinders; *C* – exhaust air from the starting valves; *D* – exhaust air into the atmosphere; *E* – lubricating oil (adapted from [10])

In the case of using a distributor of this type on reversible engines, the intermediate disk is installed movably in the housing and can be rotated at a certain angle corresponding to the amount of change in the air supply phases to start the engine in the opposite direction. A pneumatic drive is used to rotate the intermediate disk, and its position is controlled using two limit switches (Fig. 8.8 *a*).

In Fig. 8.9 shows the device (Fig. 8.9 *a*) and the principle of operation (Fig. 8.9 *b*, *c*, *d*, *e*) of an air distributor with cylindrical spools.

For preliminary review

The starting air distributor has a spool-type design with precision-machined interchangeable spool pairs located radially relative to the axis of rotation of the drive shaft. The distributor spools are controlled by a cam shaft driven by the engine camshaft. In the absence of pressure in the starting system, the spools are held by springs in the upper position, in which the starting valves connected to the spool pairs communicate with the atmosphere (Fig. 8.9 *c*). When the main starting valve opens, the spool pistons are pressed against the cam, causing the control piston for the engine cylinder that is in the starting position to pass control air to the starting valve (Fig. 8.9 *d*). The starting valve opens and supplies compressed air to the engine cylinder, causing the crankshaft to turn. As a result, the distributor cam shaft is also rotated, opening the air supply to the starting valve of the next cylinder. In this case, the previous starting valve is connected to the atmosphere through a spool pair. While the main starting valve is open, the sequential opening of the spools will be repeated until the crankshaft speed rises to the level necessary for self-ignition of the fuel. After the main start valve closes, the pressure drops quickly and springs lift the spools away from the cam. Thus, the spools interact with the cam only during the starting cycle and therefore wear is negligible.

To reverse the engine, the cam shaft is equipped with two profiles – forward or reverse. Profiles are changed by axial movement of the shaft using a pneumatic (hydraulic) drive. A cylinder is used as an actuator, in which a ledge moves, performing the function of a piston dividing the working space of the cylinder into two volumes. The supply of air (oil) to the corresponding cavity leads to axial movement of the cam shaft (Fig. 8.9 *b*).

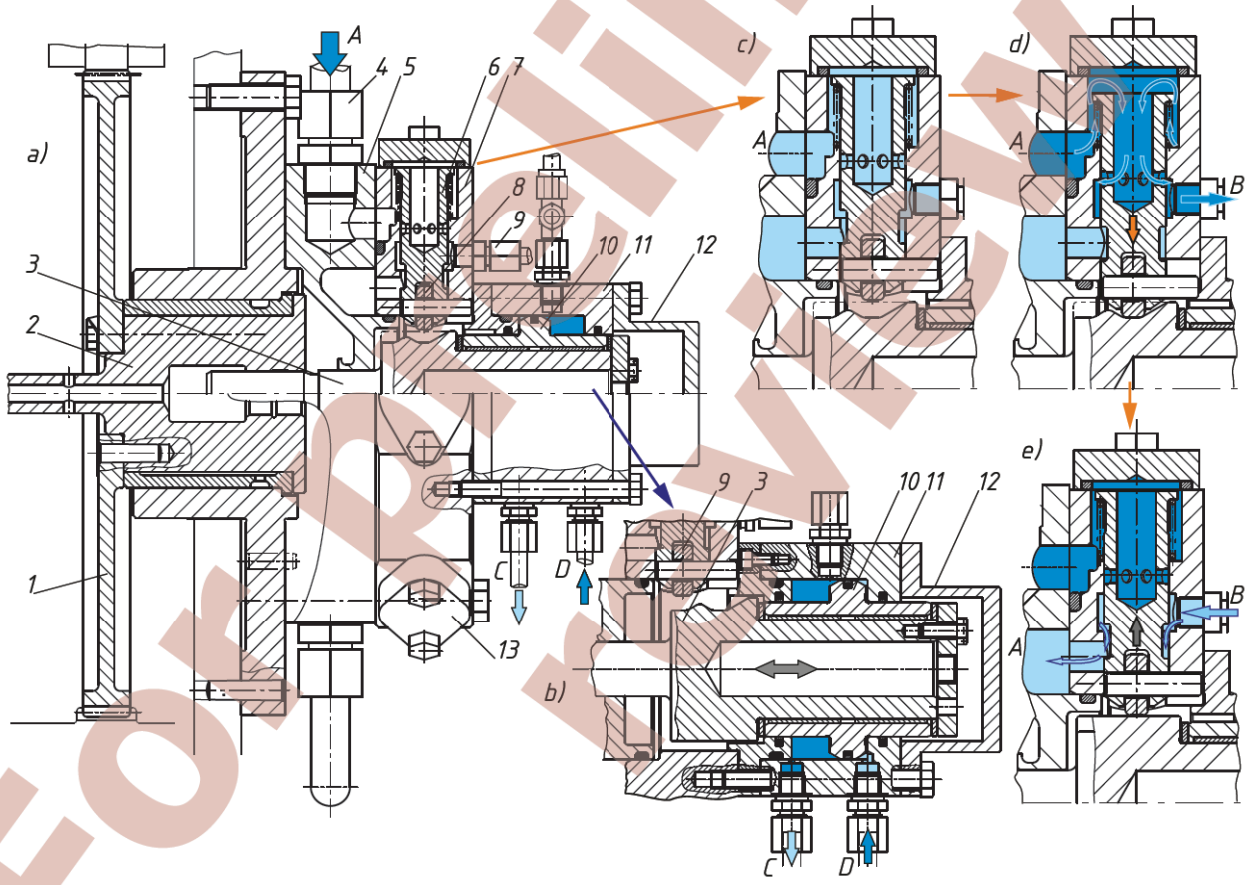


Figure 8.9 – Spool distributor of compressed air across the cylinders of Wärtsilä RTA series engines; general device (*a*), mechanism for changing profiles of distributor cams when reversing the engine (*b*); flow diagram (*c*, *d*, *e*). 1 – drive gear; 2 – hub; 3 – drive shaft with a double set of cams; 4 – air supply fitting from the main starting valve; 5 – distributor housing; 6 – spool; 7 – spool bushing; 8 – roller pusher; 9 – air supply fitting to the starting valves; 10 – piston for axial movement of the shaft; 11 – cylinder for axial movement of the shaft; 12 – cylinder cover; 13 – spool module cover *A* – air supply from the main starting valve; *B* – air to the starting valves of the working cylinders; *C*, *D* – air to the piston drive for axial movement of the shaft (adapted from [11])

For preliminary review

The starting valve serves to supply starting air to the working cylinder of the engine. In Fig. 8.10 shows the design of starting valves for low-speed (Fig. 8.10 *a*) and medium-speed (Fig. 8.10 *b*) engines.

The start valve is installed on the cylinder cover. It is driven by control air from the starting distributor. When the corresponding engine cylinder is rotated to the starting position, air flows into the piston cavity of the starting valve drive, which leads to its opening. Air from the power line enters the working cylinder. After the working cylinder leaves the starting position, the cavity of the starting valve power cylinder communicates with the atmosphere through the distributor, which leads to its closure. In the absence of control air supply from the distributor, the start valve is held closed by a spring.

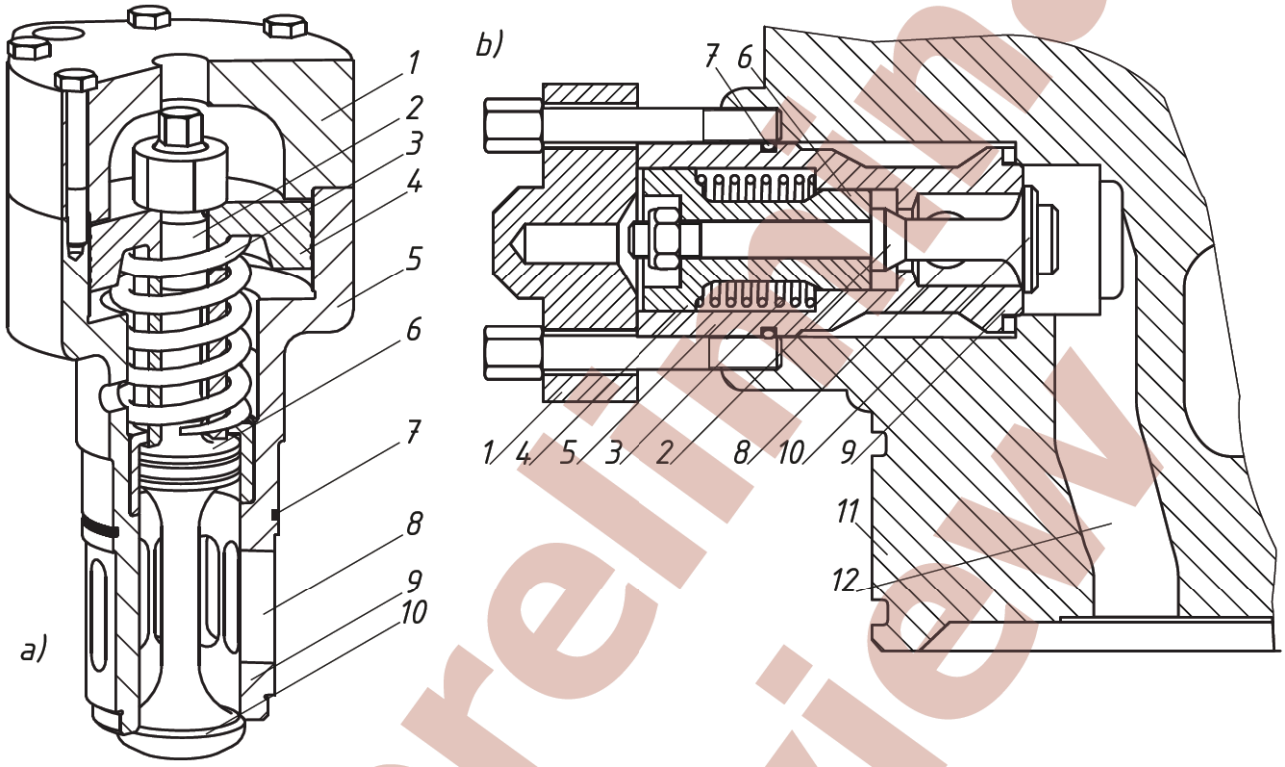


Figure 8.10 – Starting valves of low-speed engines of the MC and MC-C series from MAN (*a*) and medium-speed engines L/V 32 from Wärtsilä (*b*): 1 – chamber cover of the pneumatic piston; 2 – spindle; 3 – return spring; 4 – pneumatic piston of the valve drive; 5 – valve housing; 6 – balancing piston; 7 – seal; 8 – holes for the passage of starting air; 9 – valve seat; 10 – start valve; 11 – engine working cylinder cover; 12 – channel for supplying starting air to the working cylinder (adapted from [10, 12])

8.5 Electronically controlled direct starting systems

Direct start systems for electronically controlled engines differ from conventional systems in that they do not have a compressed air distributor, and its function is performed by individual control valves with an electromagnetic drive. Control of the start and reverse process is entrusted to the electronic engine control unit, which generates electrical signals supplied to the corresponding control valves.

In Fig. 8.11 shows a diagram of the starting and reversing system of the main engines of the X series from WinGD.

Compressed air from the receivers is supplied to the main starting valve, which at the time of start-up ensures its supply to the distribution manifold of the power drive, from where it is supplied through large-diameter pipes through a flame arrester to the engine cylinder covers.

A flame arrester in the system is necessary in case the start valve leaks (burns out or freezes). It should prevent open flame from entering the main manifold. Since compressed air always contains vapors and microparticles of lubricating oil, their ignition in an oxygen-rich environment can lead to an explosion and destruction of elements of the starting system.

For preliminary review

Air is supplied to the starting valves through channels in the cylinder covers. In this case, part of the air is diverted to the control valve installed on the piston cover of the power drive of the starting valve. In the absence of an electrical signal, the control valve connects the cavity of the power cylinder to the atmosphere, and the start valve is closed. When a signal is received, the control valve opens and supplies air to the drive power cylinder, which leads to the opening of the starting valve and supply of air to the engine working cylinder.

The starting air distribution manifold and the main starting valve are equipped with system pressure controls and condensate drain valves. Additionally, the system is equipped with a safety valve to protect it from excessive pressure.

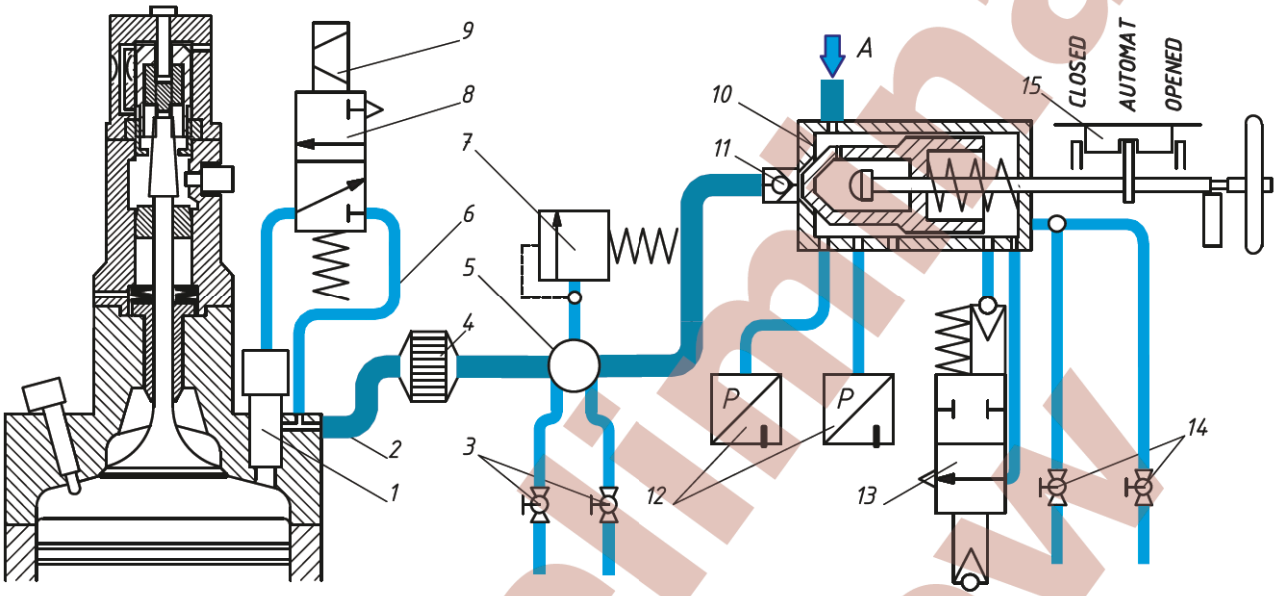


Figure 8.11 – Diagram of the air-starting system of low-speed engines of the WX series from WinGD: 1 – starting valve; 2 – air supply line to the starting valve; 3 – shut-off devices for draining condensate from the air manifold; 4 – flame arrester; 5 – starting air collector; 6 – air exhaust line to the control valve; 7 – safety valve; 8 – control valve for air supply to the cylinder; 9 – electromagnetic drive of the control valve; 10 – main air valve; 11 – check valve; 12 – means of monitoring leaks in the main valve; 13 – electropneumatic valve for controlling the air supply to the starting air manifold; 14 – shut-off devices for draining condensate from the main valve; A – air from the receiver (adapted from [13])

The main starting valve is equipped with a remote control system when it operates in automatic mode (Fig. 8.12 *a*).

The valve spindle, equipped with a handwheel, has three positions. When the spindle is fully screwed in, the valve is closed. This position is set when the engine is stopped for a long time or when work is carried out inside it. When the spindle is fully turned out, the valve is forced to open. This position is used to start the engine if the automatic control system fails.

In standby mode or during operation, the shut-off valve is in a position that corresponds to automatic regulation, and a special lever locks the spindle in this position. In this case, the starting air enters through the inlet pipe into the inlet chamber. Through the drainage hole, the space behind the drive piston of the main shut-off element is filled. When the pressures in the piston cavity and the inlet chamber are equal, the locking element is held closed by spring force.

When an electrical signal is supplied from the engine control unit to the electromagnetic drive of the control valve, air enters under the piston of the control valve drive (Fig. 8.12 *b*). By moving, the piston opens the pressure relief valve from the cavity under the drive piston of the main shut-off device. In this case, the air flow through the drainage holes does not have time to compensate for its leakage through the open control valve, as a result of which the pressure in the sub-piston chamber begins to decrease. The resulting pressure difference overcomes the spring force and opens the main starting valve, providing air supply to the starting air distribution manifold (Fig. 8.12 *c*).

For preliminary review

When the control signal from the control valve is turned off, its drive piston is connected to the atmosphere, the pressure relief valve closes, which leads to the restoration of equal pressure in the inlet cavity and the sub-piston chamber of the main shut-off device. The main release valve closes under the force of the spring.

A check valve is built into the housing of the main starting valve, which prevents the reverse movement of air from the distribution manifold into the starting valve (Fig. 8.12 *a*).

The starting valve is installed on the cylinder cover (Fig. 8.13 *a*) and has a design similar to valves of other types of engines (Fig. 8.13 *c*). The only distinctive feature is the installation of a control valve with an electromagnetic drive on its cover (Fig. 8.13 *b*). Activation of this valve by the electronic engine control unit causes air to be supplied to the cavity of the power piston and the start valve opens.

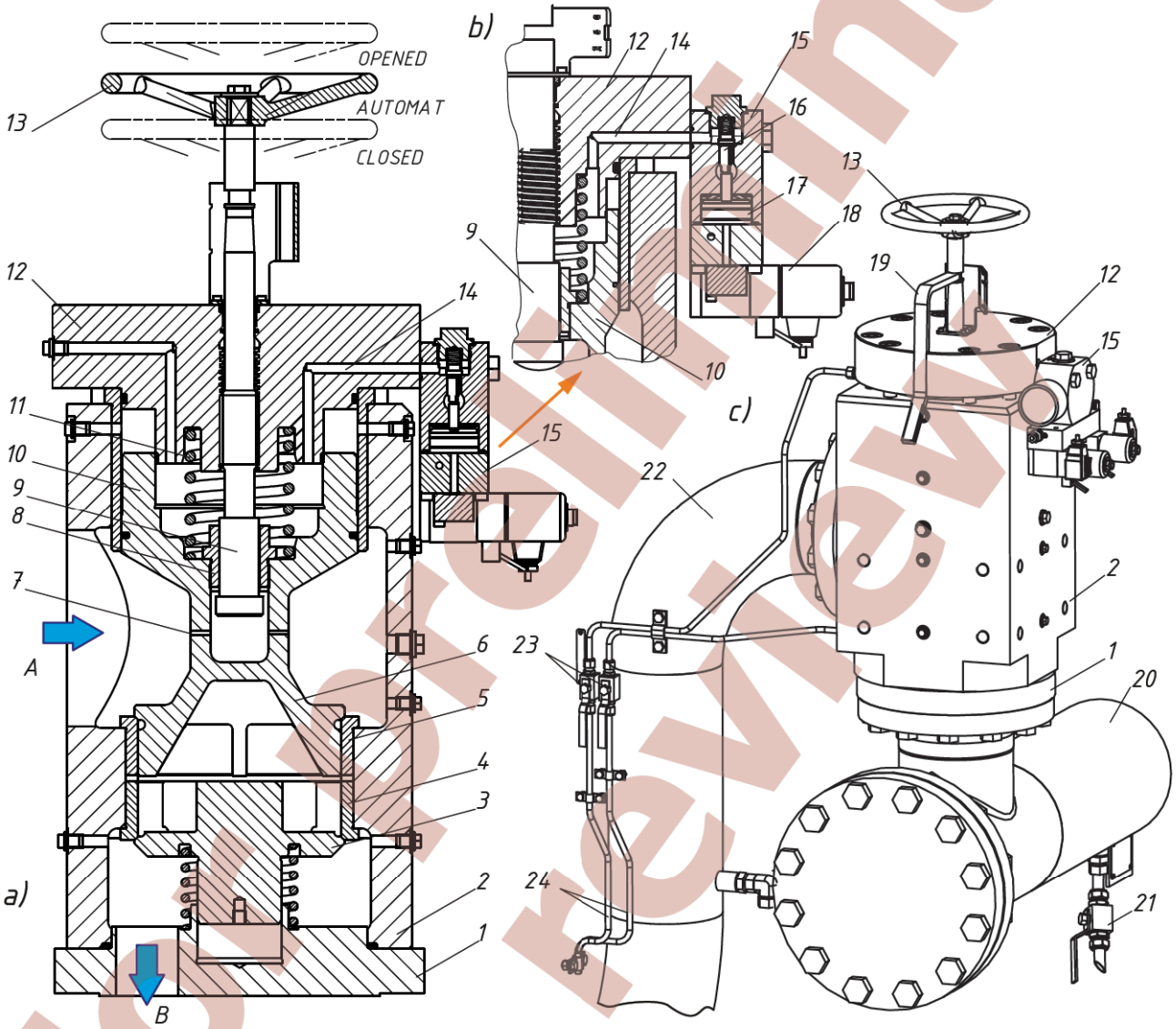


Figure 8.12 – Main air valve (*a*), main air valve control module (*b*) and general view of the main air valve (*c*) of low-speed engines of the WX series from WinGD: 1 – bottom cover; 2 – main valve housing; 3 – check valve; 4 – check valve seat; 5 – shut-off valve seat; 6 – shut-off part of the main valve; 7 – drainage holes; 8 – spindle fixation sleeve; 9 – main valve drive spindle; 10 – pneumatic piston of the main valve drive; 11 – return spring; 12 – top cover; 13 – flywheel; 14 – air exhaust channel to the control valve; 15 – electro-pneumatic control valve; 16 – pressure relief valve in the control piston chamber; 17 – pneumatic piston of the control valve drive; 18 – electromagnetic drive of the control valve; 19 – lever for fixing the spindle of the main starting valve; 20 – main starting air manifold; 21 – valve for draining condensate from the collector; 22 – compressed air supply line to the main starting valve; 23 – ball valves for draining condensate from the starting valve; 24 – condensate drain pipes; *A* – air from the receiver; *B* – air into the main manifold (adapted from [14])

For preliminary review

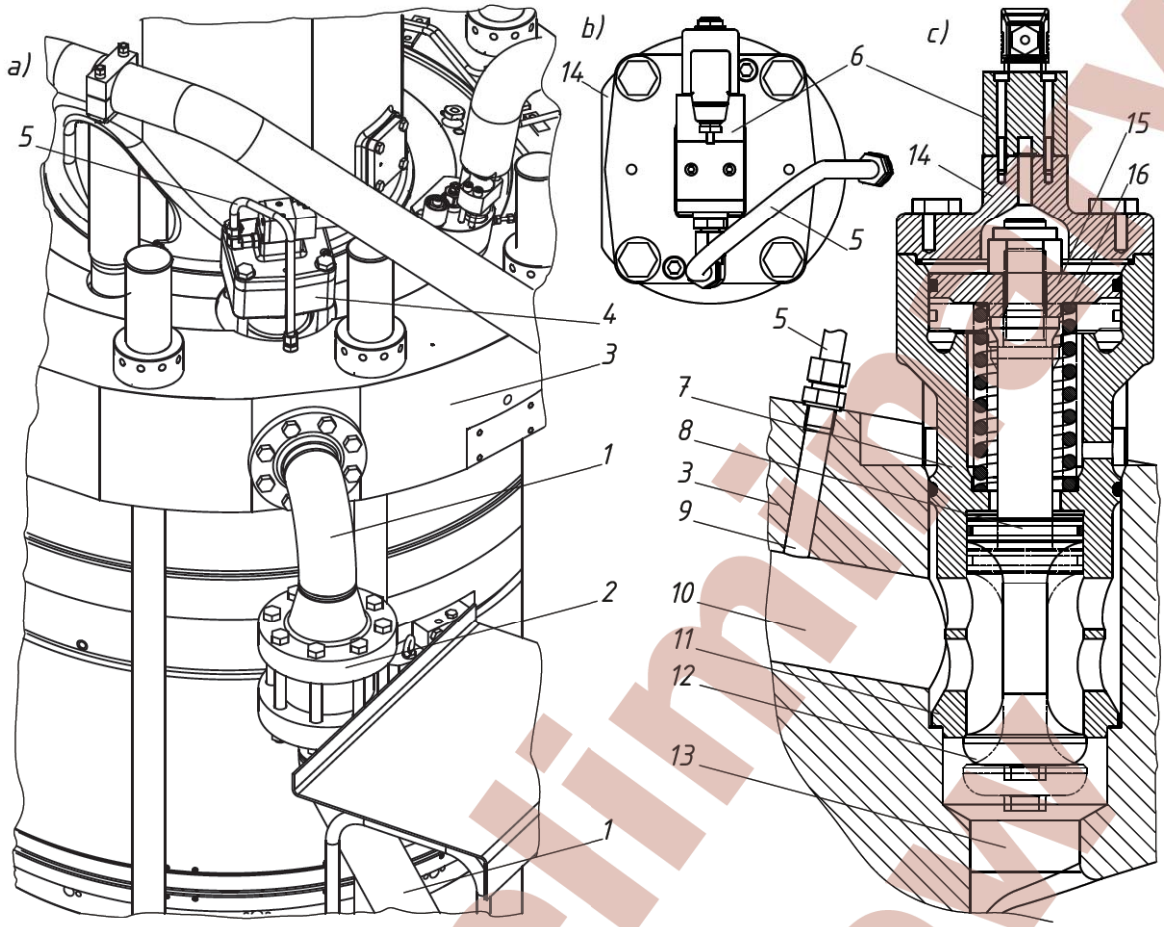


Figure 8.13 – Location of the starting valve (*a*), control valve (*b*) and starting valve (*c*) of low-speed engines of the WX series from WinGD: 1 – air supply line to the starting valve; 2 – flame arrester; 3 – cylinder cover; 4 – start valve; 5 – air supply tube to the control valve; 6 – solenoid valve for controlling the air supply to the cylinder; 7 – starting valve housing; 8 – spindle balancing piston; 9 – air supply channel to the control valve; 10 – air supply channel to the starting valve; 11 – starting valve seat; 12 – starting valve plate; 13 – air supply channel to the working cylinder; 14 – cover; 15 – pneumatic piston for starting valve drive; 16 – return spring (adapted from [14])

Literature used when writing section 8:

1. Engine Starting Systems. Ingersoll Rand. ENG32760. 2013. – 64 p.
2. Engine Starting. Installation & Operating instructions. IPU Group. 2011. – 28 p.
3. Air starters for marine applications. Turbotwin. Dayton, OH, AN99-134 2011. – 32 p.
4. Sarter Motors: AZE; AZF; AZG; AZJ. Iskra Avtoelektrika d.d. – 20 p.
5. MAN B&W L16/24. Compressed Air System. 51309-05H., Edition 07H., 08028-0D/H5250. Holeby Generatinc Sets, 1994. – 3 p.
6. Air and Gas Powered Starter Series 150BMG. Installation and Maintenance Information. P6031. 03524832. Edition 13, Ingersoll Rand, 2014. – 16 p.
7. Turbine Powered Starters Series ST600. Installation and Maintenance Information. 04662995. Edition 6, Ingersoll Rand, 2014. – 12 p.
8. Turbine Powered Starters Series ST700. Installation and Maintenance Information. P6921. 03537651. Edition 15, Ingersoll Rand, 2014. – 52 p.
9. Engine type W46B Instruction Manual. Wartsila Finland Oy Vaasa Factory. Finland. 2011. – 458 p.
10. Instructions for Main Engine. Type S46MC-C. Volume III-Component sdescriptions, Edition 2. MAN B&W Diesel A/S. Denmark. – 168 p.

For preliminary review

11. RTA72U-B. Operating Manual «Marine». Wärtsilä Switzerland Ltd, Switzerland. 2000. – 277 p.
12. Wartsila W38B Instruction Manual. Wartsila Italia S.p.A., Trieste, Italy. – 296 p. (8.10)
13. Wärtsilä W-X92. Operation Manual «Marine». Document ID: DBAC873183. Winterthur Gas & Diesel Ltd., Wärtsilä Services Switzerland Ltd. Winterthur, Switzerland. 2015. – 484 p.
14. Wärtsilä W-X92. Maintenance Manual. «Marine». Document ID: DBAC873183. Winterthur Gas & Diesel Ltd., Wärtsilä Services Switzerland Ltd. Winterthur, Switzerland. 2015. – 484 p.

For preliminary review

List of abbreviations

BDC	–	Bottom Dead Center
B&W	–	Burmeister & Wain
CASS	–	Combustion Air Saturation System
CCAI	–	Calculated Carbon Aromaticity Index
CG	–	Center of Gravity
CL	–	Controlled Leakage
CN	–	Cetane Number
CPG	–	Cylinder-Piston Group
CRISD	–	Common Rail Injection System Driver
DF	–	Dual-Fuel
DME	–	Dimethyl Ether
ECA	–	Emission Control Areas
EGR		Exhaust Gas Recirculation
ELFI	–	Electronic Fuel Injection
FAME	–	Fatty Acid Methyl Ester
FCT	–	Flex Cam Technology
FLV	–	Flow Limiting Valves
FQS	–	Fuel Quality Setting
GI	–	Gas Injector
GWP	–	Global Warming Potential
HAM	–	Humid Air Motor
HEUI	–	Hydraulically-actuated Electronic Unit Injection
HFO	–	Heavy Fuel Oil
HSE	–	High Speed Engine
HT	–	High Temperature
ICE	–	Internal Combustion Engine
ICU	–	Injection Control Unit
IFO	–	Intermediate Fuel Oil
IMO	–	International Maritime Organization
ISO	–	International Standardization Organization
K-GET	–	Kawasaki-Green Eco Turbine
LNG	–	Liquefied Natural Gas
LPG	–	Liquefied Petroleum Gas
LSE	–	Low Speed Engine
LT	–	Low Temperature
MARPOL	–	International Convention for the Prevention of Pollution from Ships
MDO	–	Marine Diesel Oil
MGO	–	Marine Gas Oil
MICE	–	Marine Internal Combustion Engine
MPC	–	Multi-Purpose Controller
MSE	–	Medium Speed Engine
PEVA	–	Proportional Exhaust Valve Actuation
PM	–	Particulate Matter
PTG	–	Power Turbine Generator
RFO	–	Residual Fuel Oil
S_aCoS_{one}	–	Safety and Control System on Engine
SCR	–	Selective Catalytic Reduction
SECA	–	Sulfur Emission Control Areas
SFWI	–	Stratified Fuel-Water Injection

For preliminary review

SIP	–	Swirl Injection Principle
STG	–	Steam Turbine Generator
TCEV	–	Top-Controlled Exhaust Valve
TDC	–	Top Dead Center
TEU	–	Twenty-foot Equivalent Unit
TFO	–	Thin Fuel Oil
ULSFO	–	Ultra-Low Sulfur Fuel Oils
VEC	–	Variable Exhaust-valve Closing
VIC	–	Variable Inlet-valve Closing
VIT	–	Variable Injection Timing
VLSFO	–	Very Low Sulfur Fuel Oils
VOC	–	Volatile Organic Compounds
VTA	–	Variable Turbine Area
VTG	–	Variable Turbine Geometry
VTI	–	Variable Turbine Inlet
WECS	–	Wärtsilä Engine Control System
WinGD	–	Winterthur Gas and Diesel
WSD	–	Wear Scar Diameter

For preliminary review

Annex 2

Brief information about scientists and engineers who made a significant contribution to the development and establishment of engine building

Huygens Christian (14.04.1629-08.07.1695) – Dutch mechanic, physicist, mathematician, astronomer and inventor. In 1678 they proposed the first piston engine, which was supposed to use black powder as fuel. The engine proposed by Huygens was never built, but the very idea of burning fuel inside the working cylinder formed the basis for the creation of modern internal combustion engines.

Nicolas Léonard Sadi Carnot (01.06.1796-24.08.1832) – son of the famous politician and mathematician Lazare Carnot. In 1812 He graduated from the Charlemagne Lyceum and entered the Polytechnic School. In 1814 was sent to an engineering school in the city of Metz, and after its completion in 1816 was assigned to an engineering regiment. In 1824 Carnot's first and only work, Reflections on the Motive Force of Fire and on Machines Capable of Developing This Force, was published. This work is considered fundamental in thermodynamics. Died in 1832 from cholera.

Sulzer-Hirzel Johann Jakob (1806-1883) and **Sulzer-Sulzer Salomon** (1809-1869) – founders of the Swiss company Sulzer Brothers. The company began its activities in 1834 from the production of fire pumps, and subsequently switched to the production of ship steam engines. Despite the fact that none of the brothers lived to see the invention of the diesel engine, the innovative spirit inherent in the founders contributed to the fact that everything new found support from the company's management. The brothers' followers realized the promise of diesel engines in time and were among the first to begin producing them for ships. For more than 80 years, the company has been one of the leaders in this industry.

Beau de Rochas Alphonse Eugène (09.04.1815-27.03.1893) French thermodynamic engineer, author of works in various branches of technology. He was involved in the laying of underwater telegraph cables, the organization of railway traffic, and the project of a flooded tunnel across the English Channel. In 1861 developed the theoretical foundations of the four-stroke cycle of an internal combustion engine and substantiated the need for pre-compression as the main condition for increasing the efficiency of the operating process. October 26 1860 received a patent for this cycle, and in 1863 published the book «Recent Research on the Practical Conditions for the Use of Heat» in Paris. Beau de Rocha limited himself only to theoretical work and did not challenge the implementation of his developments by other inventors.

Reithmann Christian (02.09.1818 – 07.01.1909) Austrian watchmaker, self-taught mechanic. For his workshop, he independently built several engines in which the four-stroke cycle was implemented for the first time. October 26 1860 he received the first patent for an engine that had a cylinder diameter 98 mm, piston stroke 111 mm and a rotation speed of 200 min^{-1}. This happened six years before Nichols Ott received a patent for his engine. And although in his patents Reitmann did not defend a four-stroke cycle, but a specific design, he subsequently managed to win a patent dispute against Otto. However, Reitmann retained all rights to the invention for Otto.

Burmeister Karl Christian (1821-1898) and **Wain William** (1819-1882) co-founded the 1846 **Hans Heinrich Baumgarten** of the engineering company, which was later renamed the company Burmeister & Wain. The company was engaged in shipbuilding, having its own shipyards and engineering factories for the production of ship equipment, including steam engines. Subsequently, the company began producing diesel engines based on a patent acquired from R. Diesel. For many years, the company remained a leader in this sector of mechanical engineering, and its research center still operates as part of MAN Diesel SE.

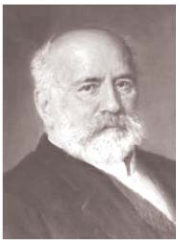
For preliminary review

Lenoir Jean Joseph Etienne (12.01.1822 – 04.08.1900). Lenoir was the first to come up with the idea of using a mixture of combustible gas and air as fuel for a piston engine. It was this idea that turned out to be decisive for the further development of internal combustion engines. In January 1860 he received a patent for such an engine. Its power was 1.5 h.p. at 1000 min^{-1}, and the efficiency was no more than 4%. The engine was a great commercial success and was used for a variety of purposes. It began to be produced by the French companies Marinoni, Lefebvre and Gaultier. About 500 Lenoir engines were built.

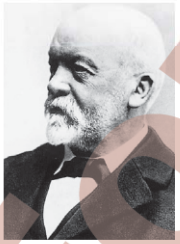
George Bailey Brayton (03.10.1830-17.12.1892) American engineer and inventor. In 1872 He patented a constant pressure internal combustion engine, the cycle of which is still used today to describe the operating processes in gas turbines (the Brayton cycle). For his engine, Brayton developed an air fuel supply system, similar to which was later used by R. Diesel, and a direct fuel supply system, an analogue of which is used in modern engines. Brayton engines were among the first to be used in transport. In 1881 such an engine was installed on the world's first submarine, the Fienian Ram.

Otto Nicolaus August (10.07.1832-26.01.1891) – German self-taught inventor. Born in Holzhausen. He studied at a real school, but did not finish it. He began his career as a salesman, and spent all his free time working on creating a gas engine of his own design. In 1864 Otto met E. Langer, who began to finance his work. Otto received a patent for his engine on April 21 1866 In January 1872 Otto and Langer founded the Gasmotorenfabric Deiz Ag campaign. for the production of gas engines, the efficiency of which reached 15%, three times greater than the efficiency of steam engines. About 5 thousand engines were produced.

Langen Karl Eugen (09.10.1833-02.10.1895) was born in Cologne, into a family of large sugar producers. There in 1850 entered the Higher Technical School at the Faculty of Chemical Engineering. In 1857 having received an engineering degree, returned to his father's factory. In August 1862 meets N. Otto and provides him with assistance in designing his engine, as well as financial support. Together they founded 1864 campaign of Gasmotorenfabric Deiz Ag, which was the first engine factory in the world.



Buz Heinrich von Ritter (17.09.1833-08.01.1918) German engineer and industrialist. He graduated from a vocational school in Augsburg. From 1851 to 1853. Studied at the Polytechnic Institute of Karlsruhe. After training, he completed an internship in Alsace and London. With 1857 begins work at the Agsburg Machine-Building Plant, and from 1864 becomes its director. In 1913 becomes CEO of the merged company Maschinenfabrik Augsburg-Nürnberg AG (now the MAN concern). He was the first to believe in the ideas of R. Diesel and provided him with all possible support, allocating funds and a laboratory for the construction of the first engines.



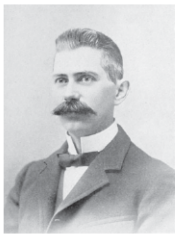
Daimler Gottlieb Wilhelm (17.03.1834-06.03.1900) – German engineer, designer and industrialist. The son of a baker from the town of Schorndorf near Stuttgart. Upon completion of primary school in 1847 began working as an apprentice gunsmith. IN 1857 Daimler enters Stuttgart Polytechnic Institute. After studying, he worked in different European countries. In 1863 he began working at the Bruderhaus factory in Reutlingen where he met Wilhelm Maybach, a capable engineer who became his business partner for the rest of his life. Together with Daimler developed one of the first Maybach cars and several types gasoline internal combustion engines.

Maybach August Wilhelm (09.02.1846-29.12.1929) – German automobile designer and entrepreneur. At the age of 13, Maybach was orphaned and lived in an orphanage for 13 years Reutlingen where he received the profession of a draftsman and designer. There, in the shelter, he meets Gottlieb Daimler, who served as the head of a machine-building plant, whose official duties included patronage of the shelter. Wilhelm Maybach was assigned to Daimler as an assistant. Maybach collaborated with Daimler in the matter of development internal combustion engine. As technical director of the Daimler Motors Society, he made a decisive contribution to the creation of the first modern automobile. Maybach is the founder of the Mercedes and Maybach automobile companies.

Preliminary review

Atkinson James (1846-24.03.1914), British engineer. In 1882, he developed a series of engines, the operating principles of which were significantly different from those previously known. To increase efficiency, the principle of reducing the compression stroke relative to the expansion stroke was proposed. Actinson developed several drive circuits to implement this principle, but they were quite complex and also led to a decrease in power density. At one time, Actinson's work did not become widely known until, in the 21st century, interest in it was revived, and the proposed method for implementing the operating processe is today known as the Atkinson cycle. Later, this method of increasing efficiency was improved and widely introduced into practice by the American engineer Ralph Miller.

Dugald Clerk (31.03.1854-12.11.1932) Scottish engineer who developed 1878 The world's first commercially successful two-stroke engine, which was patented in England in 1881. Graduated from Anderson University in Glasgow. To carry out the engine's working cycle, two separate cylinders were used, however, unlike the Brayton engine, the air cylinder was not used to compress air, but as a purge pump to clean the working cylinder of combustion products formed during the combustion process. This scheme was subsequently used for a long time on two-stroke marine diesel engines, until the purge cylinders were replaced by turbocharger units.

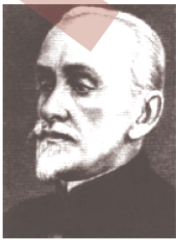


Buick David Dunbar (17.09.1854-05.03.1929) American inventor. Born in Arbroath, Scotland, he moved to Detroit at the age of two when his parents migrated to the United States. In the 1890s, Buick developed an interest in internal combustion engines. In 1899, he organized a company to produce gasoline engines for industrial purposes. And the very next year he built a car with a single-cylinder water-cooled engine. On May 19, 1903, he reorganized his company into the Buick Motor Company. In July 1904, the company introduced the Buick B car to the market, with a two-cylinder overhead valve engine with a displacement 2.6 l and power of 22 h.p. Buick's own designs.

Depp Georgy Filippovich (17.11.1854-21.03.1921) in 1877 He graduated from the technical department of the Stuttgart Polytechnic Institute and studied at the Polytechnic Institutes of Munich and Dresden. Having handed over in 1885 exam for the degree of industrial engineer at the St. Petersburg Institute of Technology, became a teacher at this educational institution. In 1898 was elected professor in the department of mechanics. On the instructions of E. Nobel, he negotiated with R. Diesel about the purchase of his patent. Under Depp's leadership, the first Russian diesel engines capable of running on unrefined oil were created. Based on the results of this work in 1901 The work «Experiments with Diesel Engines» was published.

Joseph Day (1855-1946) trained as an engineer at the School of Practical Engineering at Crystal Palace in London. In 1889, while working on an engine design that would not infringe on Otto's four-stroke engine patents, he developed his own design for a two-stroke engine that used the sub-piston space as a scavenge pump. The engine was equipped with two automatic valves, one on the intake and one in the piston crown. Later, one of Day's collaborators, Frederick Cock, proposed a design in which the piston skirt controlled the intake of fresh charge, which made it possible to completely eliminate the valves. As a result, this design became widespread and has reached our time almost unchanged.

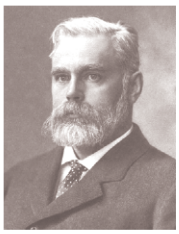
Johann Jakob Sulzer-Imhoof (30.091855-06.01.1922) – son of the founder the company of the Sulzer brothers, headquartered in Winterthur. As a result of negotiations in 1898 the company led by Joan acquires a patent for the construction of R. Diesel's engine. Joan had not only a business relationship with Diesel, but also a close friendship. The company's engineers made a huge contribution to the development of the design of diesel engines for marine use. In 1905, the Sulzer company released a two-stroke diesel engine with a valve-slot purge circuit. Later, this circuit was replaced by a slotted circuit, which made it possible to significantly simplify the design. The company produced such engines until 1982.



Brix Fedor Aleksandrovich (02.02.1855-1936) graduated from the Marine Engineering School (1887) and the mechanical department of the Nikolaev Maritime Academy (1882). Professor, specialist in the field of theory and design of ship steam engines and propulsors. He developed a method for graphic-analytical study of the operating processes of steam engines with a crank drive mechanism. Subsequently, this method, known as the Brix method, was widely used in the study of operating processes in internal combustion engines.

For preliminary review

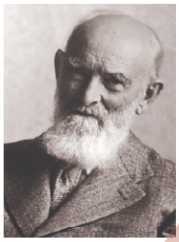
Diesel Rudolf Christian Karl (18.03.1858-29.09.1913) born into a family of German immigrants in the suburbs of Paris. In 1870 the family moved to England, from where Rudolf's parents sent him to study in Germany – first in Augsburg, and then at the Technical High School of Munich, from which he graduated with honors 1880 After receiving his education, Diesel works in the Paris branch of the Carl Linde company and at the same time is developing a highly efficient compression ignition heat engine. In 1890 he presented a theoretical basis for his idea, and in 1892 received the first patent. In 1897, a 25 hp engine was demonstrated. The new engine was of interest to the Krupp company, the Augsburg engineering plant and many others. The inventor allegedly died on September 29, 1913 under unclear circumstances: having set off from Antwerp to London on board the «Dresden» ferry, Diesel mysteriously disappeared along the way.



Nobel Emmanuel (22.06.1859-31.05.1932) – the son of the famous oil industrialist Ludwig Nobel – highly appreciated the prospects of R. Diesel's invention. One of the main motives for purchas-ing the right to produce diesel engines was his desire to increase sales of his petroleum fuel in Russia. Through Professor G. Depp, a former classmate of Diesel, Nobel agreed to sell him a license and a set of drawings for a kerosene-powered engine for 50 thousand pounds sterling. Over the course of a year, engineers at the Ludwig Nobel plant in St. Petersburg redesigned the engine to run on unrefined oil. The construction and testing of the engine was carried out under the guidance of Professor G. Depp, invited for this purpose.

Junkers Hugo (03.02.1859-03.02.1935) – professor, outstanding German entrepreneur, engineer, inventor and aircraft designer, founder of the Junkers & Co campaign, author of a number of inventions in various fields of technology. He made a major contribution to the development of the design of diesel engines with opposite-moving pistons. For refusing to cooperate with the fascist government, he spent the rest of his life under house arrest near Munich. To hide the fact that the engineering elite refused to cooperate with the Nazis, after Junkers' death a magnificent state funeral was held.

Knudsen Ivar Peter Bagger (01.04.1861-23.03.1920) – Danish engineer, director of the mechanical engi-neering department of Burmeister & Wain. He was one of the first to understand the prospects that the use of diesel engines opened up for shipbuilding. He myself visited R. Diesel in Agsburg several times and closely followed his work. On his recommendation, the company in 1898 acquired a patent for the production and further improvement of diesel engines. At Nudsen's initiative, diesel engines were designed for the cargo-passenger ship «Zealandia», which became the world's first ocean-going motor ship.



Bosch Robert (23.09.1861-12.03.1942) – German industrialist, engineer and inventor; founder of the Robert Bosch GmbH campaign. With 1869 By 1876 Studied at the Ulm School. In 1894 attended lectures at the Technical High School Stuttgart. 15th of November 1886 opened his own company in Stuttgart. In 1922 began developing a compressorless diesel fuel injection system. Already in the middle 1923 engine tests were carried out on various high-pressure fuel pumps, and in 1927 they were mass produced. With 1931 Bosch produces fuel injection pumps along with speed controllers. Robert Bosch solved the injection problem by creating a compact hydraulic system instead of a bulky compressor.

Akroyd-Stuart Herbert (28.01.1864-19.02.1927) – English engineer. Educated at Newbury Grammar School, Berkshire, and Fensbury Technical College. In 1990 received a patent for an engine with forced ignition of fuel from a preheated wall of the combustion chamber, which was later called a calorific engine. In these engines, the principle of separated mixture formation was implemented for the first time, since the combustion chamber was a spherical cavity connected to the working cylinder by a special channel. In addition, this engine was the first to use separate supply of air and fuel to the working cylinder, which was injected into the combustion chamber using a direct-acting fuel pump.

Guldner Hugo (18.07.1866-12.03.1926) graduated from the Royal Trade School of South Westphalia. With 1897 engaged in the design and manufacture of internal combustion engines. In this area, he received 12 patents, including the method of separated mixture formation. IN 1899 moves to Augsburg and for three years works as chief engineer for R. Diesel. In 1904 Together with Karl von Lidne and Georg von Krauss, he establishes the production of engines for various purposes. Known as the author of the most comprehensive guide to the history and design of internal combustion engines at that time.

Preliminary review

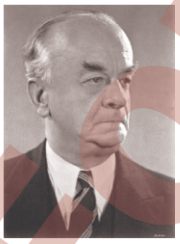
Rudolf Pawlikowski (16.06.1868-10.11.1942) German engineer. In 1893 graduated from the Dresden Polytechnic University, after which he worked with H. Junkers and O. Miller. With 1897 worked together with R. Diesel on his «rational» engine. In 1898 he moved to Dessau, where he worked as chief engineer at Görlitzer Maschinenbau AG, and in 1902 took up a position as a civil engineer in Görlitz. The main work was devoted to the technical implementation of an engine running on coal dust. In total, more than 200 patents were received. The first workable engine was built in 1916, which the inventor, by analogy with a diesel engine, called the Rupa-motor (RuPa).

Charles Yale Knight (24.02.1868-05.10.1940) – American self-taught engineer born in state of Indiana. Journalist by training. In 1901, he purchased a car from the already bankrupt company Searchmont. Soon the car's valve spring broke, giving Knight pause. In 1903-1905, he built and tested an experimental four-stroke internal combustion engine, in which gas distribution was not controlled by valves, but by a concentric pair of movable liners inserted into the working cylinder. Knight engines had a number of advantages over valves. Valveless engines had very large intake and exhaust ports, which improved gas exchange. These features made it possible to obtain high powers for those times on Knight engines.

Lanchester Frederick (23.10.1868-08.04.1946) – English engineer and inventor who made a significant contribution to the development of various branches of technology, including internal combustion engines. He studied at a number of leading technical educational institutions in England, but never received a full education. He was one of the founders of the automobile manufacturing company Lanchester Motor Company, Ltd. In the field of internal combustion engines, he developed a surface carburetor (1895), a torsional vibration damper (1910), a method of balancing a single-cylinder engine (1911), etc.

Grinevetsky Vasily Ignatievich (02.06.1871-31.03.1919) was born in Kiev, from where the family moved to Kazan, where he graduated from 1889 real school. In 1889 becomes a student at the Imperial Moscow Technical School (IMTU), which he graduates from 1896 and stays to teach. In 1902 Grinevetsky became a professor. IN 1914 he is appointed director of IMTU. Grinevetsky was one of the first in Russia to understand the prom-ise of internal combustion engines and began to seriously study them. In 1907 The first edition of G. Guldner's book «Gas, Oil and Other Engines» was published. In the appendix to the translation of the book, Grinevetsky included his work «Thermal Calculation of the Work Process», which laid the foundations for theoretical approaches to the design of internal com-bustion engines.

Zeiliger Miron Pavlovich (1874-1952) – heating engineer at the Ludwig Nobel plant. He received his education at the St. Petersburg Institute of Technology, and then taught there. There is very little information about Zeiliger's work in Russia, as well as information about himself. After the revolution, in 1924 he immigrated to Paris, where he worked as a professor at the Russian Higher Technical Institute and lectured on internal combustion engines and thermodynamics. He was a member of the board of the Society of Russian Engineers, where he lectured. The main works on calculating the operating process of high-speed engines are devoted to comparing the thermal cycles of mixed heat supply and the isobaric cycle of Diesel.



L'Orange Prosper (01.02.1876-30.07.1939) – German engineer and inventor, studied at the Technical University of Berlin Charlottenburg. After graduation he worked at Gasmotorenfabrik Deutz, and then 1910 rhe joined Benz & Cie, where he headed the production of stationary engines. In 1908 patented a high-pressure fuel injection pump with spool-type control of cyclic flow and a needle valve for closed-type injectors, and in 1909 he proposed the use of a preliminary chamber (pre-chamber) in diesel engines, which made it possible to create compact high-speed engines for ground transport. Together with his sons, he founded the L'Orange company in 1933, which today remains the world leader in the production of fuel equipment for diesel and gasoline engines.

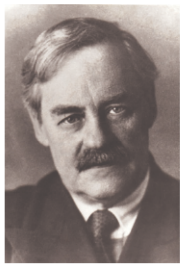
Trinkler Gustav Vasilievich (24.04.1876-04.02.1957) – Russian scientist and inventor, creator of a compressorless engine, the first design of which he presented in 1899 as my graduation project. In the spring 1902 engine production was established at the Kerting brothers plant in Hannover. Upon returning to Russia, Trinkler spent a number of years working at the Sormovo plant, improving the design of his engine. Since autumn 1917 began his teaching career. In 1930 without defending a dissertation, he is awarded the academic degree of Doctor of Technical Sciences. Until the end of his life he maintained close contact with the Sormovo plant.

For preliminary review

Sabathé Louis Gaston (03.04.1877-07.05.1949) French engineer, in 1908 submitted an application, and in 1910 for the first time patented an engine with a system for supplying fuel to the working cylinder in two stages. The first portion for supplying heat at a constant volume, the second at constant pressure. Thus, Sabathé became the first to purposefully implement in practice a cycle with a mixed heat supply and two sequential injections. There is very little information about Sabathé's life; from reliable information it is known that he worked for the thermal engine company Société des Moteurs Sabathé in the city of Saint-Etienne, which produced engines of his design for the needs of the French fleet.

Hesselman Jonas (09.04.1877-20.12.1957) – Swedish engineer. From 1899 to 1916. worked as chief engineer of company AB Diesel Engines in a suburb of Stockholm. With 1916 created his own company. Under his leadership in 1925 Multi-fuel engines with direct fuel injection into the combustion chamber were developed, combining the advantages of diesel engines and Otto engines. He made a great contribution to the study of mixture formation processes in engines with direct fuel injection. The combustion chamber shape he proposed is still widely used in modern diesel engines.

Buechi Alfred (07.11.1879-10.27.1959) – Swiss engineer, graduated from the Technical University of Zurich. He worked as head of the advanced development department of the Sulzer brothers. IN 1905 received a patent for a combined heat engine, which combined a piston engine, a centrifugal compressor and an axial turbine located on the same shaft. Subsequently, in 1908 , due to the difficulties of coordinating all the elements of the combined engine in terms of speed, the gas turbine and compressor were separated into a separate unit called a turbocharger. In 1926 founded the Buechi syndicate, the first specialized company for the production of turbochargers.



Masing Evgeniy Karlovich (16.04.1880-24.03.1944) graduated from 1906 Moscow Higher Technical School and began working there in the laboratory of internal combustion engines. With 1920 professor, head of department. Developing the ideas of his teacher V.I. Grinevetsky, improved the method of thermal calculation of internal combustion engines, studied the issues of gas generation and its use in engines. Masing's works on fuel combustion formed the basis of modern methods for designing internal combustion engines.

Nusselt Wilhelm (25.11.1882-01.09.1957) – German engineer. He graduated from the Technical University of Munich in 1904, and already in 1907 defended his doctoral dissertation on the topic «Thermal conductivity of insulating materials». In 1915 Nusselt published his pioneering work, Fundamental Laws of Heat Transfer, in which he first proposed dimensionless groups, now known as the fundamental parameters in similarity theory of heat transfer. The calculation methods proposed by Nusselt are still widely used in the study and modeling of operating processes in internal combustion engines.

Ricardo Harry (26.01.1885-18.05.1974) – English engineer and inventor. Graduated from a number of educational institutions, including Cambridge University. He became interested in the design of internal combustion engines and founded a successful company for their design and production, Ricardo Consulting Engineers Ltd. The company invested huge amounts of money in research work, thanks to which it is still a leader in the design and development of engines for various purposes. Riccardo and the staff of his campaign made many inventions in the field of internal combustion engines, including vortex chamber mixture formation, liner gas distribution and much more.

Vasily Trofimovich Tsvetkov (19.01.1887-01.12.1954) – an heating scientist, graduated with the Kharkov Institute of Technology in 1911 and came to the Kharkov Locomotive Plant, where he immediately became involved in the development of a two-stroke diesel engine with a power of 900 hp. From 1928 to 1932 g. – Technical Director the Kharkov Locomotive Plant. IN 1921 organizes and heads the Department of Heat Engines at KhPI, which in 1930 was reorganized into the Department of Internal Combustion Engines. In the same year, he became the first head of the department of aircraft engines at the Kharkov Aviation Institute. In 1940 he was awarded the academic degree of Doctor of Technical Sciences.

For preliminary review

Clessie Lyle Cummins (27.12.1888 – 17.08.1968) – founder of Cummins Engine Co. Became the first American to build and install diesel engines in trucks, buses and cars. He received more than 30 patents for inventions, including many fuel injection systems for his engines. By age 12, Cummins had built a working steam engine. In 1919, he founded the Cummins Engine Company and began producing marine diesel engines. Ten years later, Cummins installed the first diesel engine in a car, which he drove for promotional purposes 800 miles to the New York Auto Show for only $1.38 in fuel. The following year, a Cummins diesel car took part in the Indianapolis 500, finishing 12th and driving the entire track without refueling.

Vanscheidt Vsevolod Aleksandrovich (07.08.1890-27.09.1982) graduated from the mechanical department of the St. Petersburg Technological Institute in 1914 Worked at the «L. Nobel» (now «Russian Diesel»). Under his leadership in 1925 designed and built the country's first 2-stroke non-compressor engine with a power of about 37 kW. With 1929 conducted teaching work at the shipbuilding department of the Leningrad Polytechnic Institute, and in 1930 1971 Head of the Department of Internal Combustion Engines at the Leningrad Shipbuilding Institute, professor. In 1938 And 1950 His textbooks on the theory of marine internal combustion engines were published.

Pielstick Gustav (25.01.1890-11.03.1961) – German engineer. Graduated from the Kiel Higher School of Shipbuilding and Mechanical Engineering. In 1911 began working at MAN as a design engineer for submarine engines. During World War I he became chief engineer, and after the war he developed four-stroke engines for commercial shipping. In 1934 become director of the MAN diesel engines department. He paid special attention to the development of high-power turbocharged engines. With 1946 moved to France, where he founded the diesel engine design bureau Société d'Etudes des Machines Thermiques (SEMT). Pielstick's work as director was so successful that the company was subsequently named SEMT Pielstick.

Wankel Felix Heinrich (13.08.1902-09.10.1988) – German self-taught inventor. IN 1924 he came up with the idea of creating a rotary piston engine, which he worked on all his life. It was only possible to build a working engine until 1957, after engineer Walter Freude joined the work. To create a functional engine, Wankel carried out extensive research and formulated requirements for effective moving seals. Despite the lack of special education, in 1969 Wankel becomes a doctor of technical sciences at the Munich Institute, and then a professor.

Vibe Ivan Ivanovich (27.12.1902-27.12.1969) – scientist, specialist in the field of internal combustion engine theory, Doctor of Technical Sciences. In 1926 Graduated from Zaporozhye Mechanical Engineering Institute. In 1932 he graduated from graduate school at the Leningrad Civil Aviation Training Center. In 1938-41.Head of the Department of Special Engines at the Stalingrad Mechanical Institute. WITH 1965 Professor of the Chelyabinsk Polytechnic Institute. Author of numerous works on the theory of internal combustion engines. The formula he proposed for calculating combustion processes has become widespread in all countries of the world.

Glagolev Nikolai Matveevich (1903-20.04.1976) scientist in the field of theory and design of internal combustion engines. From 1954 to 1970 Head of the department of internal combustion engines at the Kharkov Polytechnic Institute. He made significant efforts to develop the scientific school of engine building in Kharkov and to its recognition both in the country and abroad. In 1949 he defended his doctoral dissertation on the topic «A new method for calculating the working processes of internal combustion engines». In 1950 he was awarded the title of professor. Author of 11 books and more than 100 works in the field of internal combustion engines.

For preliminary review

Marks:

For preliminary review

https://sites.google.com/view/modern-marine-engines/

Навчальне видання

БІЛОУСОВ Євген Вікторович
БУЛГАКОВ Микола Петрович

Двигуни внутрішнього згоряння сучасних морських суден

Підручник для вищих морських навчальних закладів

Англійською мовою

Всі права захищено. Книга, жодна її частина не можуть бути відтворені в будь-якій формі без письмового дозволу власників авторських прав.

Підготовлено в редакційно-видавничій фірмі «Прес-кур'єр»
Свідоцтво про внесення видавця до Державного реєстру видавців, виготівників і розповсюджувачів видавничої продукції серія ДК № 3764 від 22.04.2010 р.
(65008, м. Одеса, пл. Б. Дерев'янка, 1, оф. 717,
тел./факс (0482) 64-96-58, E-mail: pk.gazeta.odessa@gmail.com).

Формат 60×84/8 Ум. друк. арк. 59,52. Наклад 300 прим. Зам. № 401

Віддруковано з готового оригінал-макета.

Видавництво і друкарня «Астропринт»

65091, м. Одеса, вул. Разумовська, 21. Тел. 7-855-855.

www.astroprint.ua e-mail: astro_print@ukr.net

Свідоцтво суб'єкта видавничої справи ДК №1373 від 28.05.2003 р.

https://sites.google.com/view/modern-marine-engines/

Internal combustion engines of modern marine vessels